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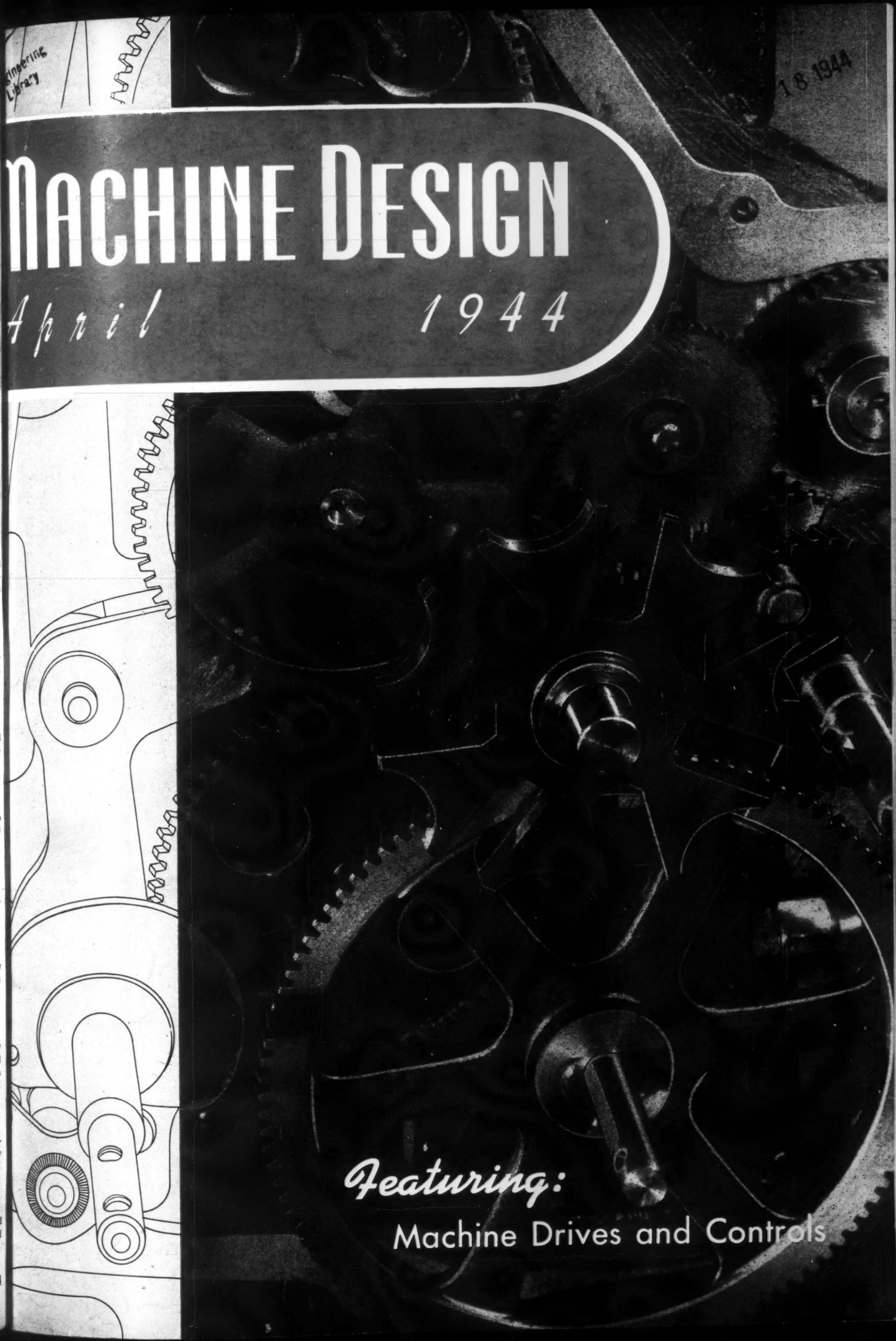
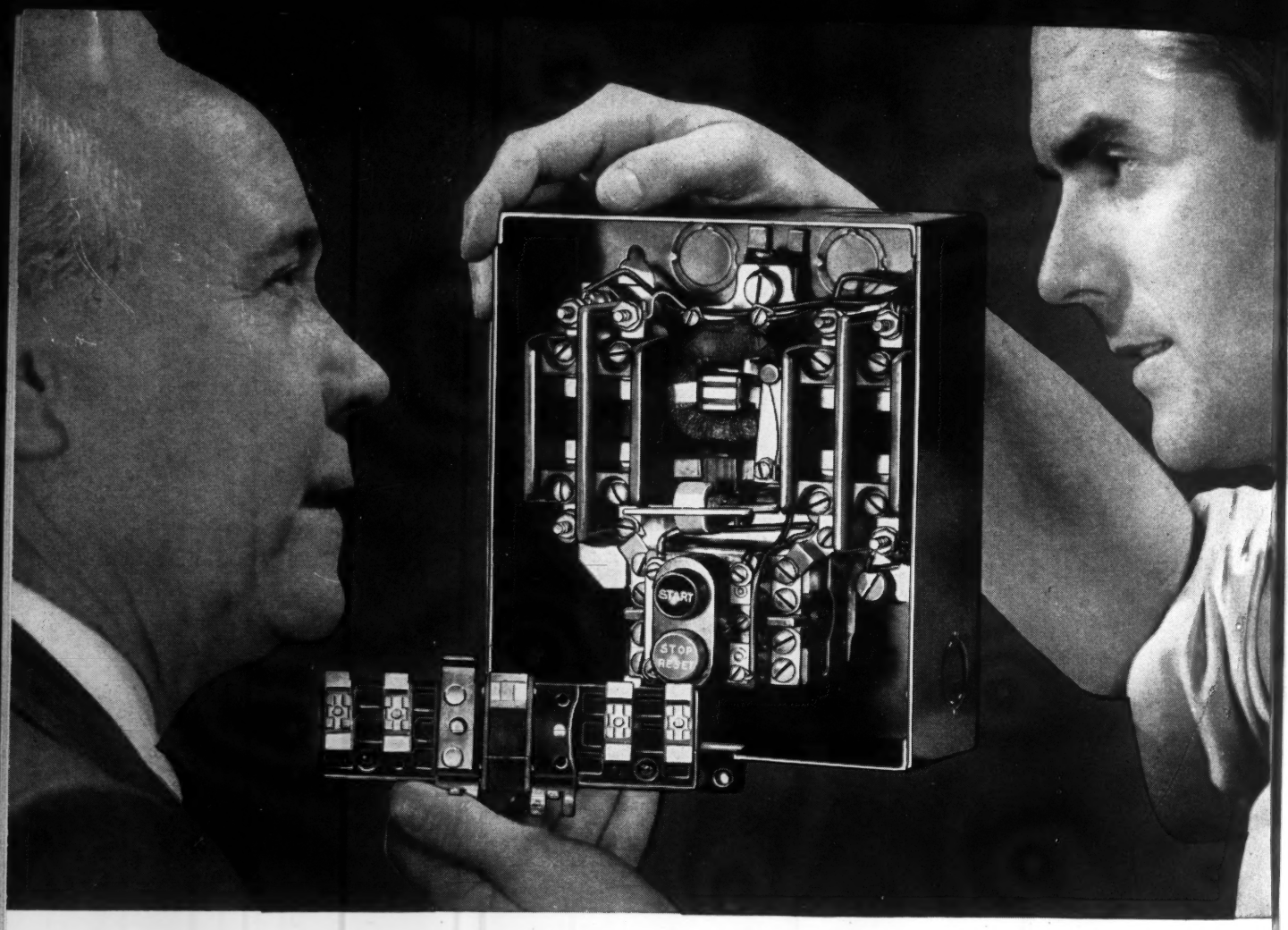
MACHINE DESIGN

April 1944



Featuring:

Machine Drives and Controls



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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

APRIL, 1944 Volume 16—Number 4

Contents

Cover—Business Machine Mechanism (Courtesy International Business Machines Corp.)

This Issue at a Glance 7

Topics 96

What's Ahead for Automotive Engines and Transmissions? 99
By Robert Cass

Engineers' Symposium 103

Basic Factors in Applying Speed Governors 111
By G. Forrest Drake

Fundamentals of Hydraulic Valve Selection 115
By William H. Stanton

Specifying Variable-Speed Drives — Part I — Mechanical 119
By Colin Carmichael

Which Small Motor To Use? 125
By C. T. Button

Built-in Automatic Temperature Controls 129
By V. G. Vaughan

Control Criterion for Hydraulics 134
By R. S. Elberty

Combined Methods Give Best Results 135
By William C. Oberem

Design Pointers in Cable Control System 141
By R. R. Richolt

Ultraspeed Camera Utilizes Constant Feed 146

Solenoids—For Power, For Control 147
By M. L. Jeffrey

Flexible Shafting Solves Remote Operation Problem 151
By George G. Eisenbeis

Electronic Control Speeds Production 154
By H. G. Grondahl

Obtaining Sensitivity with Air-Hydraulic System 155
By F. A. Barnes and C. Johnson

Electric Auxiliary Drives for Aircraft 159
By Richard M. Mock

Need Is for Increased Speeds More Automatic Control (Editorial) 163

Outstanding Designs 164

Scanning the Field for Ideas 170

Choosing the Right Material — Part V — Reliability and Probability 172
By H. W. Gillett

Selecting V-Belt Drives for Maximum Economy (Data Sheet) 175

Applications — of Engineering Parts, Materials and Processes 178

Stainless Steels AISI Types 410 and 416 (Materials Work Sheet) 179

Assets to a Bookcase 183

Design Abstracts 184

Men of Machines 185

Noteworthy Patents 188

New Parts, Materials and Equipment 190

Business Announcements 202

Calendar of Meetings 206

New Machines 208

Helpful Literature 345

For Itemized Table of Contents See Page 5

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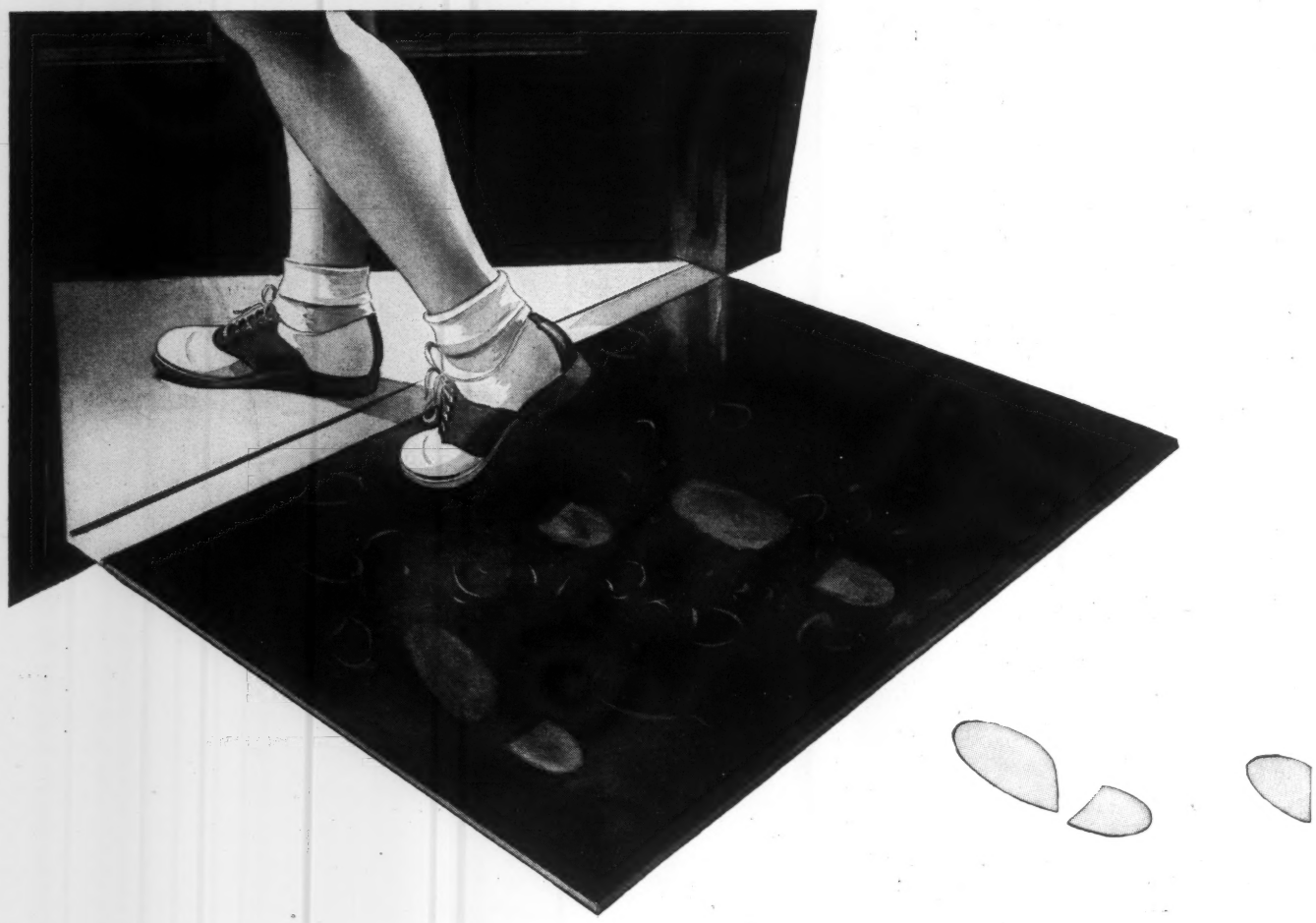

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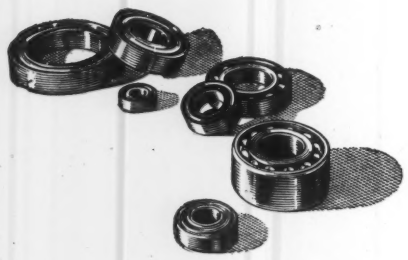

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Itemized Index

Classified for Convenience when Studying Specific Design Problems

Design Calculations:

V-belt drives, selecting for economy, Edit. 175-177

Design Problems:

Air-hydraulic system provides sensitivity, Edit. 155-158
Auxiliary motor drives for aircraft, Edit. 159-162
Built-in slip clutch simplifies motor drive, Edit. 106-107
Cable controls, design pointers for, Edit. 141-145
Control criterion for hydraulics, Edit. 134
Coordinating variable speed units, Edit. 110
Cycle timers, selection of, Edit. 103-105
Designing hydraulically controlled machine, Edit. 135-140
Engines and transmissions, designing for future, Edit. 99-102
Flexible shafting for remote drives, Edit. 151-153
Governors, basic factors for applying, Edit. 111-114
Helical cam improves hydraulic valve, Edit. 107-108
Hydraulic valves, fundamentals of selection, Edit. 115-118
Obtaining precision temperature control, Edit. 129-133
Optical system obviates shutter, Edit. 146
Precision control for processing, Edit. 105-106
Shaft drives on rotating drum facilitated with V-belt, Edit. 109-110
Small motors, where to apply, Edit. 125-128
Toggle linkage simplifies brake design, Edit. 108-109
Variable speed drives, specifying, Edit. 119-124

Engineering Department:

Design service, Adv. 227, 254, 319, 330, 364
Equipment and supplies, Adv. 28, 68, 88, 214, 298, 310, 368
Instruments, Adv. 353

Finishes:

Paints, Adv. 44, 45

Materials:

Aluminum alloys, Adv. 247, 275, 358
Bronze, Adv. 207, 343
Felt, Adv. 244, 266
Glass, Edit. 198; Adv. 64
Indium, Adv. 366
Magnesium alloys, Adv. 210, 264
Molybdenum, Edit. 184; Adv. 201
Nickel alloys, Adv. 79, 193
Plastics, Edit. 184; Adv. 40, 59, 86, 225, 262, 263, 291, 333
Powder metal, Adv. 199
Rubber and synthetics, Edit. 200; Adv. 10
Steel alloys, Edit. 172-174, 179-182; Adv. 41, 42, 43, 249 356, 365
Tungsten carbide, Adv. 295
Zinc, Adv. 194

Parts:

Balls, Adv. 358
Bearings, Edit. 101; Adv. BC, 4, 11, 14, 19, 39, 69, 76, 187, 197, 211, 213, 226, 231, 258, 296, 316, 322, 344
Bellows, Adv. 74, 283, 311
Belts, Edit. 106, 110, 175-177; Adv. 288, 315, 369
Blowers, Adv. 362
Brakes, Edit. 108; Adv. 330
Brushes, Adv. 360
Cable controls, Edit. 141-145
Cams, Edit. 110, 137; Adv. 320
Carbon parts, Adv. 24, 303
Cast parts, Adv. 8, 46, 209, 235, 278
Chains, Edit. 110; Adv. 12, 13, 30, 37, 84, 268
Chucks, Adv. 375
Clamps, Adv. 366
Clutches, Edit. 106; Adv. 94, 316, 327, 334, 340
Compressors, Adv. 318
Controls (electrical), Edit. 104, 105, 108, 129-133, 147-150, 154, 159-162; Adv. IFC, 54, 58, 62, 63, 69, 89, 91, 95, 189, 190, 192, 195, 196, 217, 221, 224, 242, 243, 276, 298, 300, 309, 323, 331, 340, 349, 355, 370
Counters, Adv. 75
Couplings, Adv. 202, 312, 337, 351, 352
Electrical accessories, Edit. 171, 198; Adv. 284, 293, 320, 329, 366
Engines, Edit. 99-102, 178, 188; Adv. 206, 316, 367
Fastenings, Edit. 198, 200; Adv. 1, 6, 16, 17, 23, 32, 36, 218, 255, 256, 296, 304, 306, 321, 324, 334, 339, 350, 361, 362, 373, 378
Filters, Adv. 31, 341
Fittings, Edit. 190, 200; Adv. 15, 237, 324, 356
Floats, Adv. 304
Forgings, Adv. 25, 232, 261, 297, 348, 360
Gears, Edit. 110; Adv. 70, 74, 92, 212, 216, 292, 306, 312, 313, 314, 332, 338, 342, 359, 371
Governors, Edit. 111-114; Adv. 22
Heating units, Edit. 200
Hoists, Adv. 318
Hose, Adv. 55, 229, 230, 302, 368
Hydraulic equipment, Edit. 103, 107, 134, 135-140, 155-158; Adv. 9, 97, 234, 250, 251, 265, 271, 279, 286, 298, 301, 308, 335
Joints, Adv. 219
Lubrication and lubricating equipment, Edit. 101, 110, 178, 188, 192; Adv. 61, 287, 236, 332
Machined parts, Adv. 245, 314, 354
Motors, Edit. 106, 125-128, 159-162, 178; Adv. IBC, 26, 29, 38, 47, 48, 53, 65, 66, 71, 72, 73, 82, 83, 93, 184, 203, 205, 233, 239, 267, 269, 272, 274, 280, 304, 326, 338, 340, 342, 356, 360, 362
Mountings (rubber), Adv. 215
Plastic moldings, Adv. 34, 35, 236, 259, 273, 282
Plugs, Adv. 299
Pneumatic equipment, Edit. 155-158; Adv. 260, 307, 325, 347, 363
Pulleys, sheaves, Edit. 141-145; Adv. 85
Pumps, Edit. 178; Adv. 27, 204, 289, 310, 312, 322, 328, 330, 338, 340, 354
Seals, packings, Adv. 2, 80, 81, 191, 248, 277, 328, 336
Shafts, flexible, Edit. 151-153, 160-162; Adv. 332, 350
Speed reducers, Edit. 190; Adv. 50, 51, 285, 300
Springs, Edit. 108, 192; Adv. 87, 270, 302, 304, 314, 320, 350, 377
Stampings, Adv. 18, 322, 342
Thermostats, Edit. 129-133, 198; Adv. 228
Transmissions (variable speed), Edit. 99-102, 110, 119 124; Adv. 49, 78, 246, 306, 352, 354
Tubing, Adv. 33, 57, 90, 222, 223, 240, 241
Universal joints, Adv. 60, 372
Valves, Edit. 107, 115-118, 170; Adv. 98, 208
Welded parts and equipment, Adv. 20, 21, 294, 310, 318, 336, 364
Wheels, casters, Adv. 308
Wire cloth, Adv. 357

Production:

Grinding, Adv. 56, 257, 358
Hardening, Adv. 52, 317
Service facilities, Adv. 252, 253, 290
Shaving, Adv. 320
Tools, Adv. 281, 334

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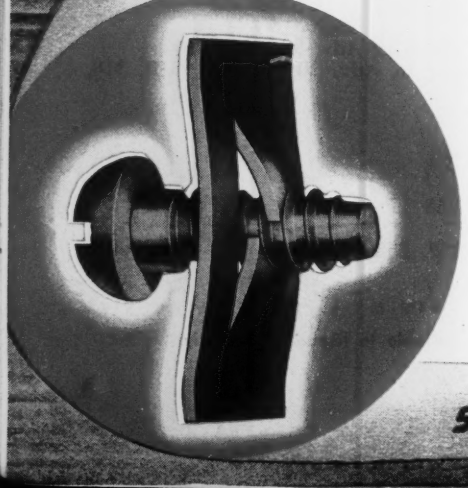

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SPEED NUTS - THE FASTEST THING IN FASTENINGS!

This issue at a glance

How Will Automobiles and Trucks of the Future . . . .

. . . . be affected by recent developments in engines, transmissions and fuels? Robert Cass discusses these significant developments and evaluates them in the light of what they might lead to after the war. Page 99.

Foolproof Control Is a "Must" . . . .

. . . . in the design of special-purpose war production machines. Without it, unseasoned operating personnel could not function at top efficiency. Just how such control was achieved by employing "combined methods" in the design of a propeller-cam milling machine makes worth-while reading. Page 135.

Aircraft Control Systems Should Be Designed . . . .

. . . . on the basis of deflection, not strength. So says R. R. Richolt in a discussion which emphasizes the importance of rigidity in holding control "springiness"—arch enemy of precision control—to a minimum. Page 141.

Keeping Abreast of Contemporary Thinking . . . .

. . . . makes it possible to sense trends—keep in step with the field. "Engineers' Symposium" presents the recent experiences of top-flight engineers in solving drive and control problems. Page 103.

Are Electromechanical Units Less Vulnerable . . . .

. . . . in fighting aircraft than hydraulics? Are they more easily installed? What are the prime factors in their design? Be sure to read Richard M. Mock's views on this vital phase of control. Page 159.

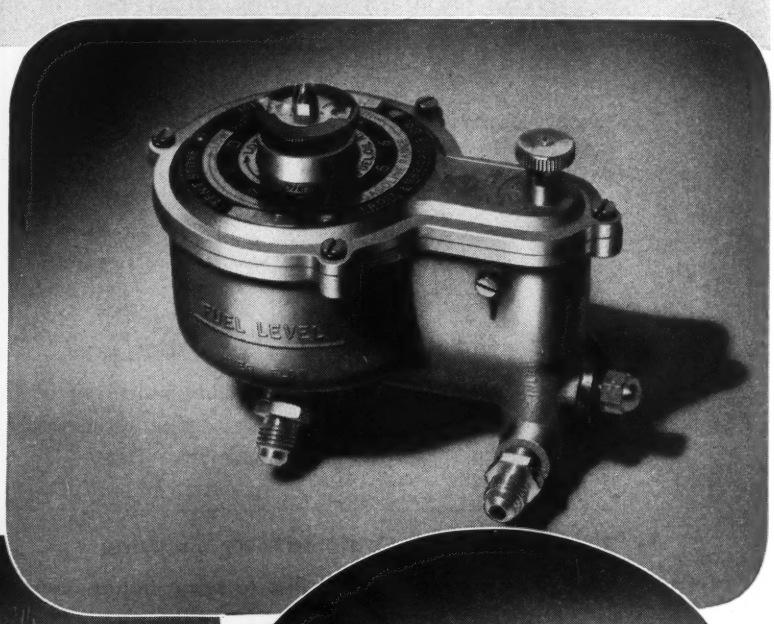
Heart of Every Machine—Drives and Controls . . . .

. . . . constitute for designers a subject of widespread appeal. Packed with up-to-the-minute data on drive and control applications, this issue of MACHINE DESIGN runs the gamut from cameras to trucks, with feature discussions on engines, motors, transmissions, governors, valves, solenoids, flexible shafting, temperature controls, etc.,—the whole comprising an up-to-date compilation of outstanding interest and value.

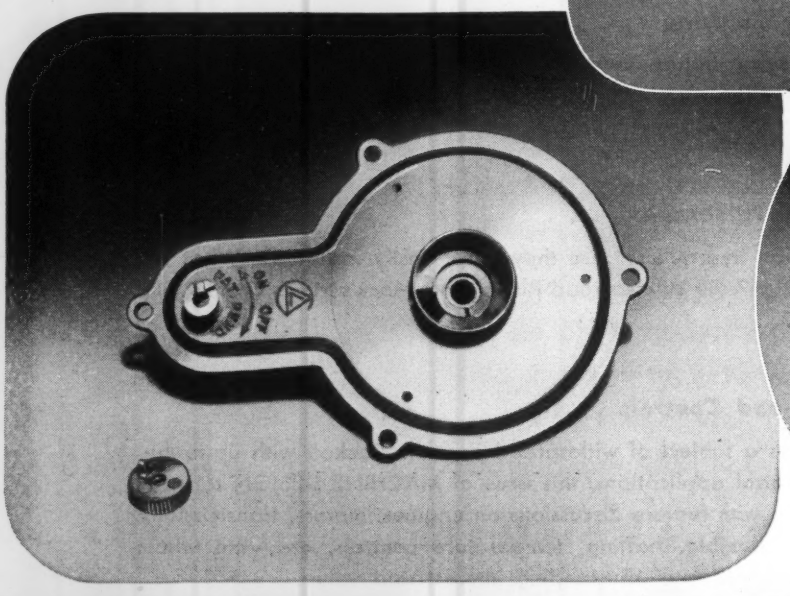
Die Castings by MADISON-KIPP

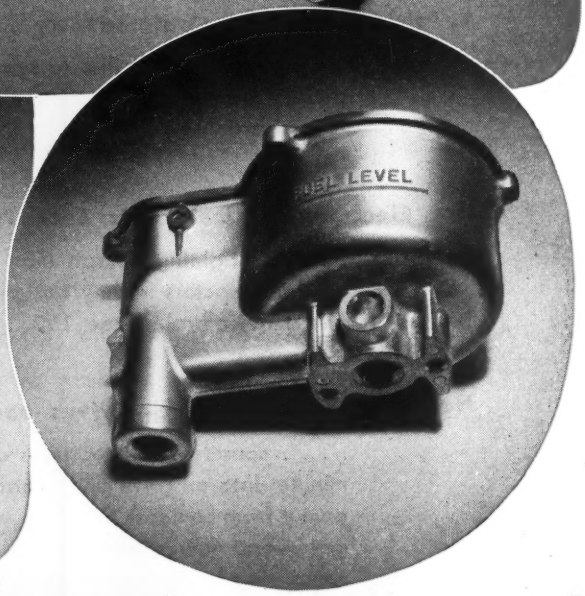
AFFORD ALMOST COMPLETE FREEDOM OF DESIGN

Engineers at Detroit Lubricator Company have long used die castings for important components of their control valves. They know the die casting process well and so from first idea sketch to finished drawing, they utilize the "free" values and economies offered by parting line coring, parting line metal savers, parting line engraving and contours. If designers know this first fundamental point of working from the parting line or lines, die casting affords great freedom for "good-looking" products at an over-all cost saving.



Another product of Detroit Lubricator Company, Detroit, Michigan, showing money-saving intricate coring and cam tracks which can be economically produced only by the die casting process.





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DESIGNER'S DIGEST

TIMELY HIGHLIGHTS ON G-E PRODUCTS

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ELECTRIC HEAT

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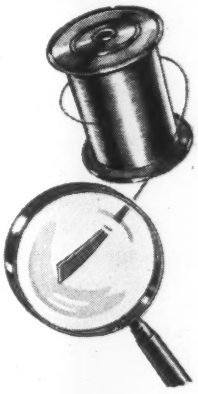

One-fourth as thick as the thinnest of previous Formex* rectangular wires, the new ultra-thin Formex offers the space savings of rectangular wire to makers of electronic and small electric components now using round wire. Smooth, strong, and well suited for high-speed winding, this new wire possesses the advantages of tough, high-dielectric-strength insulation, and inherently high space factors which have made Formex magnet wire famous. Bulletin GEA-3911.

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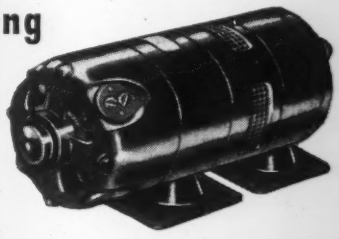
Precise Positioning at a Finger's Touch



If you're seeking a practical way to make machines more automatic you'll find that the G-E amplidyne opens up many interesting possibilities. On a boring mill, for example, it helps provide automatic boring-head positioning that's accurate to .002 inch in 10 feet of travel. On power shovels, hoists, and other equipment this swift, powerful amplifier serves as a vital link between the operator's touch and the precise motion of heavy loads.

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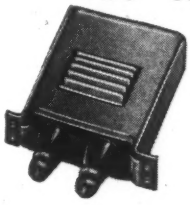

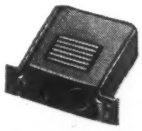

Whatever your requirements, the chances are you'll find the answer in G-E's standard line of Pyranol* capacitors for built-in applications. They combine high capacitance per cubic inch with high resistance to extremes of temperature, vibration, and humidity, characteristics that are permanent and uniform. Many of the more than 350 ratings are available in oval, cylindrical, or rectangular shapes. Cat. GEA-640.

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Topics

LITERALLY A FLYING GUN PLATFORM, the new Northrop P-61 Black Widow has been called the first functional night fighter of the war. Tests have shown that nothing short of a pillbox can withstand one blast of its battery of guns. Designed to develop pursuit speed, it retains the easy flying characteristics and low landing speed necessary for night operation.

STEEL CASTINGS industry turned out about 26¾ million individual castings during the last quarter of 1943, according to an estimate of the Steel Division of WPB, based on an average weight of 50 pounds per casting. Actually 669,000 tons of steel castings were fabricated in that quarter. Total production, for comparison for the past three years was 1,577,736 tons, 2,239,693 tons and 2,742,604 tons for 1941, 1942 and 1943, respectively.

BRITISH FIRMS have to a certain extent avoided the need for intensive instruction in reading machine drawings by issuing drawings in pictorial form. Three advantages attributed to pictorial views are: Saving of time, reduction of mistakes, smoother flow of work. Because perspective freehand sketches may be executed rapidly, single-purpose sketches are utilized widely to impart instructions which normally would be incorporated in a complete working drawing.

TINY AMOUNTS of hydrogen in steel, even as small as one two-thousandth of one per cent by weight, can make steel brittle and should be avoided in making metal for important ware uses where ductility is essential. While the effect is evident in ordinary steel, it is exaggerated in most steels containing manganese, according to Dr. Uhlig, metallurgist for General Electric.

ELECTRONIC MASTER MIND, employing 126 vacuum tubes and designed by Philco to calibrate Signal Corps radio apparatus, obviates painstaking and laborious hand operations subject to human error. Consisting of three parts, the calibrator supplies a source of standard frequencies, a means of mechanically driving the dial of the frequency meter, and a method for electrically generating a sharp pulse every time the meter is turned through "zero beat" with the standard signal.

SUPPLIES OF COPPER and copper-base alloys are sufficiently favorable, according to W.P.B., to permit revocation of an order which previously had prohibited the use of this metal for civilian purposes in automobile radiators, gaskets, fuel and oil lines, brake and clutch lining, rivets and other essential parts.

MORE WPB RESTRICTIONS are being eased monthly; for instance, restrictions on the distribution, sale and use of secondary aluminum and hard-facing materials—alloys containing cobalt, chromium, tungsten, nickel, molybdenum or vanadium—have been removed due to the satisfactory supply-demand situation of the materials involved. On the other hand, the quantity of platinum needed will exceed production, requiring that no change in control be made at this time.

FILTERINC MATERIAL, known as multipore, is produced from natural rubber latex and contains as many as 6400 perforations to the square inch. Useful in many industries these filtering screens developed by the U. S. Rubber Co., are made of either hard or soft rubber and may be compounded to resist abrasion, high temperatures, and alkaline and acid solutions as well as certain oils and greases.

INDUSTRY in the United States is producing more finished products for war and essential civilian purposes in proportion to available raw materials than it was prior to the establishment of the Controlled Materials Plan in November, 1942. Distributing materials to industry in an orderly manner permits high efficiency in the use of available resources. For example, from the inception of CMP to the end of 1943, production of airplanes increased from 14,380 to 25,953 planes per quarter, merchant vessels from 2,924,000 to 5,295,000 deadweight tons and combatant ships from 156,506 to 559,220 tons.

STEEL PROPELLER BLADES have assisted in giving the P-47 Thunderbolt thirty per cent faster rate of climb, according to Curtiss-Wright Corp. With these hollow steel blades the Thunderbolt now can outclimb the best German fighter planes while still retaining the already existent edge in top speed.

What's Ahead for Automotive Engines and Transmissions?

By Robert Cass
Chief Engineer
White Motor Company

BECAUSE of war requirements which call for a degree of mechanization greater than had been conceived in any prewar thinking, engineers naturally are interested in possible developments in engines, transmissions and other means of supplying and applying power to passenger and commercial vehicles. The development of tracks for Army use is a typical case in that it caused the stirring of minds with regard to applying tracks to vehicles hitherto using wheels. Also, much has been said regarding the technological development of fuels and it is felt in some

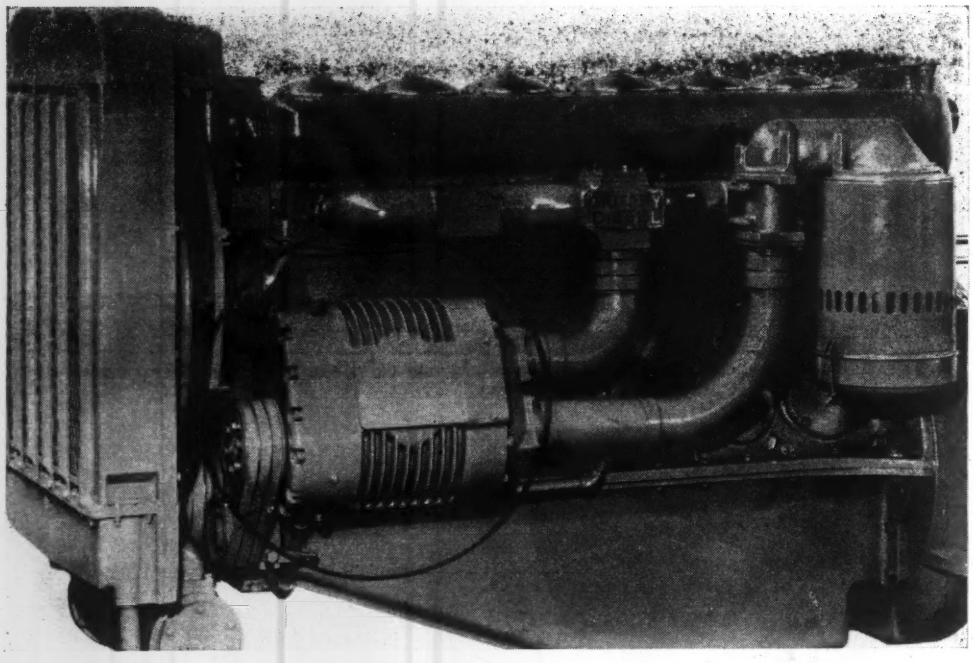


Fig. 1—Left—Supercharger, shown encircled, offers the most promising means of increasing the horsepower of automotive diesels

quarters that in the postwar era we may emerge with fuels which will permit startling changes in the economics of vehicle transportation.

In this discussion it is proposed to deal primarily with truck transportation units, although many of the same possible trends could be applied just as readily to passenger cars. It is necessary first, however, that some assumptions be made in connection with future developments. Number one assumption is that our economy in the postwar world will be a free one and not a regulated or planned one. Under a planned economy, which of course means government-regulated economy, we might easily have standards which would restrict future development. In assuming that free competition will prevail, all the advantages that have accrued to such a system in the past may be expected. A second assumption is that a significant increase in octane fuels will be available a few years after the cessation of hostilities. Statements made in this article therefore must be considered as applying to several years after the war—the intervening time prior to possible new developments of the character described being filled with existing types of vehicles with minimum changes. Now, however, is the time to consider possible long-range plans.

Some careful work has been done in analyzing the possibilities of fuel improvements—an outstanding example of which is the recent S.A.E. paper by A. T. Colwell on the effect of wartime fuel developments on postwar cars, in which it was predicted that octane ratings immediately after the war would be 75 to 77 for regular and 85 to 87 for premium gasolines. While this analysis is probably accurate with respect to passenger cars, it is not impossible, in fact it is believed probable, that developments in engines can be made for trucks and busses which will call for the use of even higher octane premium fuels. Even though these fuels will carry a slightly increased cost the overall performance, particularly in truck operation, will justify their use on economic grounds. It should be stated at this stage that fuels as high as 100 octane are not contemplated, but it may be that something in excess of 90 (A.S.T.M.) will be available for the highway transportation vehicles of this country. This will then call for designs of engines with much higher brake mean effective pressure than hitherto. In fact, compression ratios might conceivably run to the neighborhood of 8 or 9 to 1, increasing the output some 25 to 30 per cent, *Fig.* 3. This output may further be boosted by means of supercharging, resulting in an engine in which the cubic inch displacement will be considerably less for the same gross weight and the same road speeds.

It will be seen that through the introduction of the supercharger for acceleration and for hill climbing, a substantial reduction in the weight per horsepower may be made with materials now commonly used. If the possibilities of aluminum are considered in connection with engine design, as well as other units, the field becomes a highly interesting one in connection with increasingly economical truck operation within state laws governing gross vehicle or axle weights.

Fuel Injection versus Carburetion

Side by side with this phase of gasoline engine development, work is being projected for the future on the substitution of fuel injection for carburetors. It is felt that, with any given octane fuel, higher compression ratios can be used with fuel injection than are possible with carburetors, and a further gain in output can be expected. Axiomatically, any major increase in power per cubic inch should call for better specific fuel consumption, and interesting speculations are being made as to the ultimate that can be achieved. Elimination of the part-load, part-throttle wastage inevitable with the usual carburetors may combine with other savings to justify the initial cost of the fuel-injection system. However, this is still in the embryonic stage.

Hand-in-hand with this desire for better engine performance must come development of valves and bearings

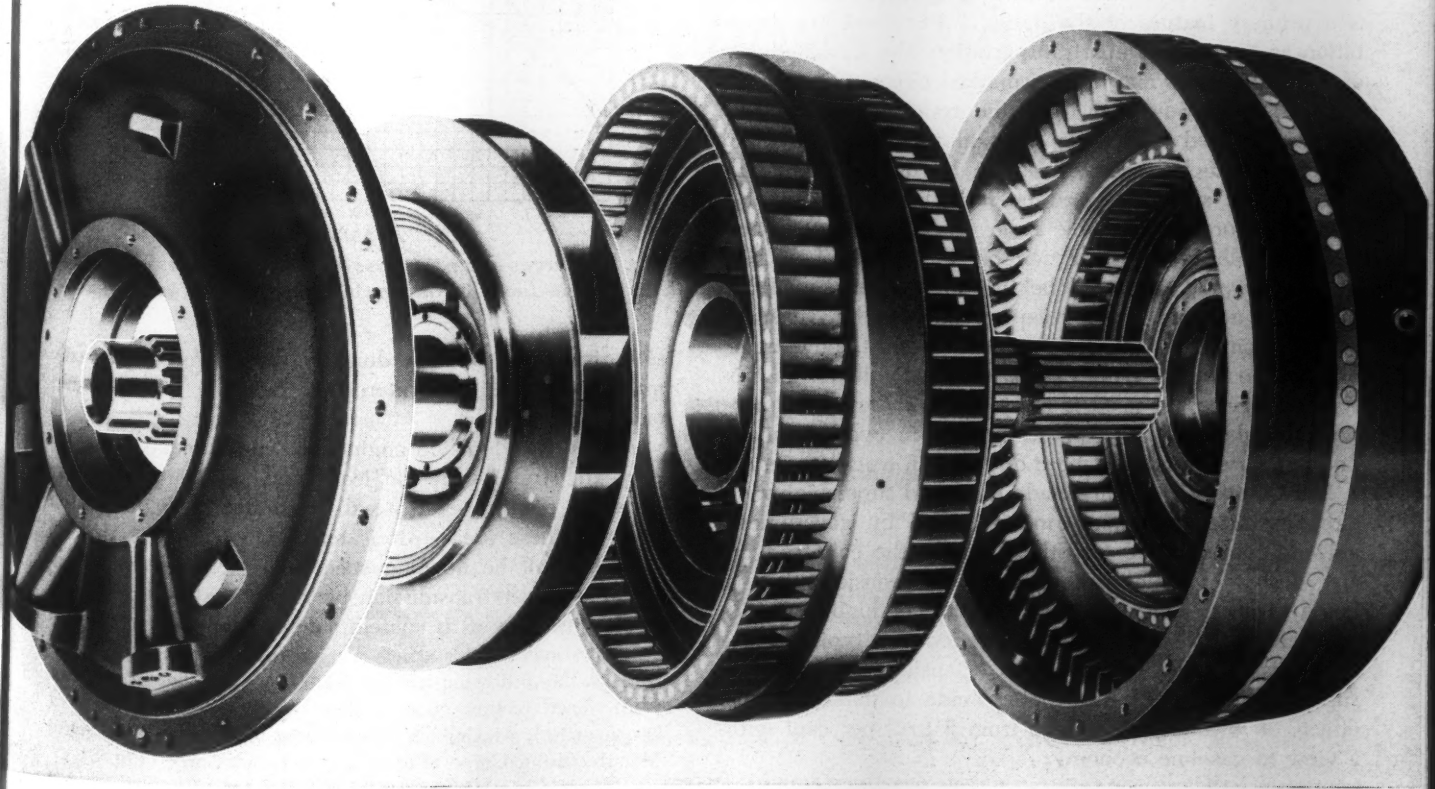
—the two components most likely to call for more intensive research. In this connection as also with pistons, an excellent background is found in aviation developments. It will, however, be necessary to produce these parts at costs far below those commonly found in aircraft manufacture if the price per horsepower is to be reduced. Certainly progress in commercial engine development can be measured only on such a basis. Currently the truck field uses engines in which the loadings on connecting rod bearings are in the neighborhood of 1200 pounds per square inch. The bearings used carry a nominal rating of 3000 pounds per square inch. It will, therefore, be necessary to develop bearings which will sustain higher loads because of the necessity of reducing engine sizes, and correspondingly reducing bearing sizes, if the overall weight reductions are to be carried to a logical conclusion. There is every indication that the bearing and valve manufacturers will be able to meet any demands that the engine designers contemplate with special fuels that are likely to be available.

A fact not usually appreciated is that the load factor in the case of an over-the-road truck engine is greater than any other engine used for commercial purposes. Vehicles pulling trailers across the highway currently have load factors in the neighborhood of 80 per cent whereas the airplane engine has around 50 per cent load factor and, of course, the passenger car very much less than this. This is an important consideration in the detail design of such engines. Within the field of truck engine sizes it seems reasonable to suppose that with the high compression ratios contemplated, it will not be vitally necessary to go to overhead-valve engines where L-head engines have been used in the past. To this extent major changes in tooling setups will not be necessary. It therefore seems that, given improved gasolines, it is reasonable to expect much smaller engines to be doing the work of highway transportation than will be possible if fuels remain at low-octane levels.

There are two other major considerations with respect to these engine changes—cooling and lubrication. Desire to reduce still further any unnecessary dead weight may well result in the adoption of higher cooling temperatures, and, therefore, pressure cooling systems may be universally used. The Army, of course, to a minor degree already has gone in that direction, but to make gains which are important it will be necessary to carry higher pressures than have been used. Naturally this calls for active consideration of water-pump designs, radiator construction, and such engine items as block distortion, block materials, etc.

Having made a step in that direction, it is natural to examine what will be the lubrication problem. Here again changes will have to be made to take care of these new conditions. Oils used today are no longer so-called straight mineral oils but are largely chemical compounds in which inhibitors and detergents are freely sprinkled, so it may be anticipated that the chemists will perform some more sleight-of-hand tricks to provide oils which will be highly satisfactory at temperatures in the neighborhood of 300 degrees both from the lubricating angle and the consumption—the miles per quart figures in good

Fig. 2—Below—Hydraulic torque converter includes a housing, shown at extreme right, provided with two sets of guide vanes which take the torque reaction



modern engines are quite high.

Up to now discussion has dealt solely with the gasoline engine, but unquestionably the diesel engine also has a definite place in our postwar economy and it would be interesting to analyze whether any startling changes have been made during the war that would seriously change the postwar picture. Up to the present the development of fuels used in diesel engines has not undergone comparable changes and, therefore, it seems now that relatively small changes in diesel fuels may be expected in the postwar. era.

Under such circumstances any increase in the horsepower for these engines would seem to lie in the direction of mechanical means such as supercharging, *Fig.* 1. It may, therefore, be assumed that the price per horsepower will not be lowered substantially due to design changes, and any reduction considered will probably come from manufacturing facilities that may have been improved during the war. With the possible advent of fuel injection for gasoline engines the specific fuel consumption will begin to approach more nearly that of the diesel engine and it seems likely that in truck-size engines where the diesel has had a fairly undisputed sway, the gasoline engine may challenge it for many operations.

Scope for Automatic Transmissions

With respect to transmissions, there is some thought that the influence of the passenger car may focus the spotlight of the engineer's attention on automatic transmissions for trucks The two general types which may be considered are the mechanical and the hydraulic. The automatic mechanical transmission includes a fluid-drive element which, however, functions merely as a shock-absorbing coupling and does not effect any torque conversion. In the hydraulic transmission, torque conversion is a primary feature of the design. For truck transportation as opposed to bus transportation, the development of super highways may have a marked effect on transmission designs. So-called hydraulic transmissions, before the war, carried with them a premium in fuel costs, so that before such transmissions can be used in commercial vehicles such increased operating expense must be eliminated or be offset by some other factors that are demonstrable in the operation of the vehicle. In bus transportation it may well be that all the large-capacity busses engaged in city service operation will have automatic transmissions, and it is anticipated that the hydraulic or torque converter type, *Fig.* 2, will predominate. In this type of service the need is most apparent but it is questionable whether they can be justified in overland main line bus service and still more doubtful in truck service.

Plans already are being made to extend super highways already started and many more backed by government aid, are contemplated for other parts of the country. In any design of super highways careful consideration is being given to grades so that first cost of highways will be secondary to economy of operation. A typical illustration is that in a contemplated extension of an existing super highway studies are being made to determine the effects of lowering the grades from 3 to 2 per cent with a view to gasoline economy.

With regard to the ultimate transmission of the power to the road, live axle designs must be considered. Here again changes are contemplated—although such changes will not be in quite the same revolutionary class as those previously mentioned. Following the work that has been done on hypoid axles in passenger cars, interest is being shown in designs for manufacturing larger hypoid axles, particularly by those who are proprietary manufacturers of such units. It may, therefore, well be that such axles will be offered to the commercial field in the future, and so long as the attendant lubrication problems do not present as serious a problem in the truck field as they did in the passenger car field, they will be given a chance to prove under actual operating conditions the claimed advantage of long life.

From time to time independent wheel springing comes to the fore for consideration in commercial vehicle design and it is not unlikely that in light-duty commercial vehicles, as well as busses, new forms of suspensions will be tried. However, the question of weight and cost will have to be favorably answered, as the ride characteristics for most commercial vehicles are not so seriously wrong as to warrant paying any penalty for a slight improvement. Nevertheless an adoption of some form of suspension rising out of work done in the aircraft field easily may prove a highly interesting field for research and experimentation.

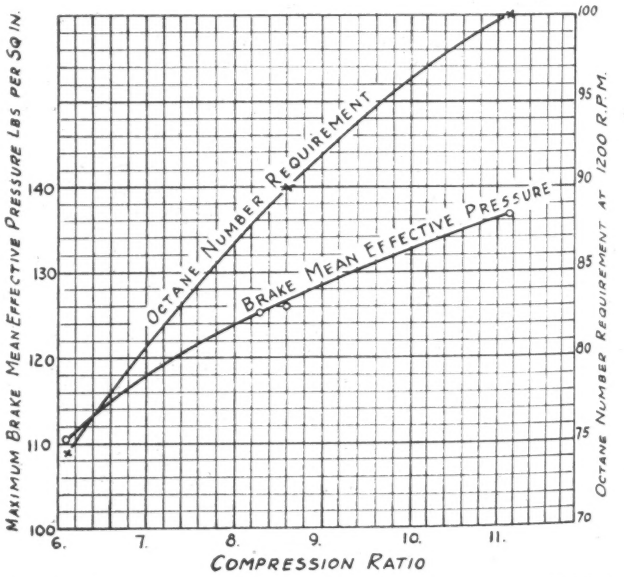


Fig. 3—Curves show the possible gain due to use of higher-octane fuels in an engine designed to use them

All the war-depleted engineering departments undoubtedly are straining at the leash and as soon as the war ends the pent-up energy will no doubt be expended on some of these ideas. It is too much to expect that all of them will be howling successes but it seems certain that the power transmission picture is destined to be just as revolutionary as is contemplated for appearance.

Illustrations for this article were furnished through the courtesy of the following companies: Bohn Aluminum & Brass Corp. (head picture of an engine of the future as conceived by an artist); McCulloch Engineering Corp. (*Fig.* 1); Spicer Manufacturing Corp. (*Fig.* 2); and Ethyl Corp. (*Fig.* 3).

This "engineers' symposium" comprises a group of articles which discuss solutions to drive and control problems encountered in the design of specific machines. Engineering soundness has been proved by the success of the applications. The necessity for optimum utilization of parts and materials in present designs as well as in plans for postwar developments makes mandatory the careful consideration of all phases of drives and controls—the heart of every machine. The articles in the symposium are presented with the knowledge that solutions to problems in one type of machine often suggest the answers to difficulties which confront designers of other machines in widely diversified fields

Synchronous Timer Assures Uniform Cycles

By Mark Morgan
Electrical Engineer
Reed-Prentice Corp.

HYDRAULICALLY operated and electrically controlled injection molding machine illustrated utilizes a hydraulic system to actuate the die plates and solenoid-operated valves to control the plunger. To design an electrical diagram for the control of a group of solenoid valves at a predetermined cycle and suitable for automatic, semiautomatic, and hand operation, is a task presenting several interesting points.

For hand operation the task of controlling the cycle of operation is left to the

judgment of the operator. The machine is equipped with two lever-operated transfer switches, one for the dies and one for the plunger. They are double-throw switches with neutral position, whereby the operator is enabled to stop die plates or plunger independently at any intermediate point between the two extremes of travel.

Three Timers Control Cycle

When automatic operation has been selected through a selector switch, the job of controlling the cycle is taken over by a group of timers. The selector switch has double-throw, sustained contacts with no neutral position. Its function is to select between single cycle and continuously repeated cycle. When selection of the type of cycle has been made, automatic cycle is started by means of a pushbutton which connects, through an auxiliary relay, the network of the timers to the network of the solenoids.

It is impossible, in a discussion of general character, to go into the details of the wiring diagram. However, it will be interesting to note how three timers are enough for the control of four valves—close and open dies, plunger in and out. In other words, the number of timers is equal to the number of operations minus one. This is accomplished by having each timer control the following operation: The timing out of the last timer will start the first operation, and the program will repeat itself indefinitely. Such an arrangement may leave a doubt in the mind of the reader regarding the starting of the first operation the first time. This is done by the "start" pushbutton, and the cycle is then picked up automatically.

The first unit controls the time the plunger stays forward to maintain pressure on the material. The second is for solidification after the plunger has returned, and the third controls the dies so that they stay open to allow for ejection of the product. When the third unit has timed out, it closes the circuit of the "close dies" valve again, thus initiating the next cycle.

Besides the three timers mentioned, the machine is equipped with a fourth timer to control the delayed unloading solenoid valve located in the hydraulic circuit of the plunger. This timer is always connected in the network regardless of whether the machine is running on hand or automatic operation, has a five-seconds adjustable dial, and is so connected in the circuit that its timing is started by a limit switch operated by the plunger when reaching the extreme "in" position. The load-contacts sequence is such that contacts close when timing begins, open when timing out, stay open when power is off. The purpose of this timer is to allow the pump's full volume to be applied to the plunger when making the shot and to hold it for a number of seconds, as called for by the timer, before the delayed unloading valve is de-energized and releases pressure in the cylinder. Incidentally, all the valves are electric remote controlled, both for automatic and hand operation.

Current Capacity Is Primary Consideration

In the choice of a suitable timer among the many types available, a matter to be given first consideration is the current-carrying capacity of the load contacts. It is known that a solenoid, when energized with its plunger all out for maximum stroke, will draw an instantaneous peak current due to the comparatively large air gap in the magnetic circuit; such gap is closed by the plunger, as it is attracted in, through action of the magnetizing force induced by the ampere-turns in the coil.

In the type of solenoid valves installed in this machine, the current ratio, maximum to minimum stroke, is approximately 10 to 1, at 220 volts, 60 cycles. It should also be remembered that the coil of an alternating-current solenoid is a highly inductive type of load and that the current will reach the peak value at the closing, not at the opening of the circuit, and that the danger or probability of freezing contacts is greater when closing than when opening a circuit.

In order not to complicate the diagram excessively and to keep the dimensions of the control panel within reason-

able limits, it was found advantageous to have some of the timers' contacts operate on the circuit of the solenoids directly; that is, without using intermediate auxiliary relays.

Another important element is the timing accuracy which should insure constantly identical repetition of a given time cycle throughout the total range of the time setting. It has been determined that a calibration of 120 seconds is a good practical value for timers applied to the control of plastic injection molding machines, and that an accuracy of one second over a 120-second cycle, that is, less than 1 per cent, can also be practically obtained. The timers installed on this machine have a 120-second maximum setting, with a vernier-type adjustable dial, and an accuracy within plus or minus .005.

Timer Must Be Sturdy

The question of space is also of paramount importance. Modern machines have all the control panels built within the housing and mounted in separate compartments. National Electric Code, Section 6732, calls for such enclosures not to contain or be connected to any other strictly mechanical or moving equipment, which also sometimes brings up the problem of space. Moreover, a certain amount of pounding is hard to prevent as it is in the nature itself of the machine. For these reasons, a timer of compact design and sturdy construction is to be given preference over the other types.

An attempt to list all the various types of timers would be uncalled for in this discussion, therefore, just a few will be mentioned: Pneumatic timers where the time element is provided by a certain amount of air flowing through an orifice restricted and adjustable by means of a pin movable within the orifice itself, mechanical timers, such as those featuring a Telechron synchronous motor which controls the coil of a solenoid clutch; and electronic timers, where a vacuum tube (pliotron or thyratron) has the grid connected in series with a capacitor and so biased to the cathode to prevent the tube from firing when the timer is off.

The control circuit of electronic timers is so arranged that when timing begins, the capacitor starts to discharge through a resistor; at end of timing, the grid voltage is such to allow the tube to fire and thus to pick up a relay and control the load. For this molding machine, synchronous motor timers were selected.

To summarize, it may be said that the timers are the most delicate and important element of an automatic plastic injection molding machine. The machine's performance is directly dependent on the performance of the timers. They are called upon to stand a heavy-duty cycle on strict specifications and often present delicate problems to designing engineers.

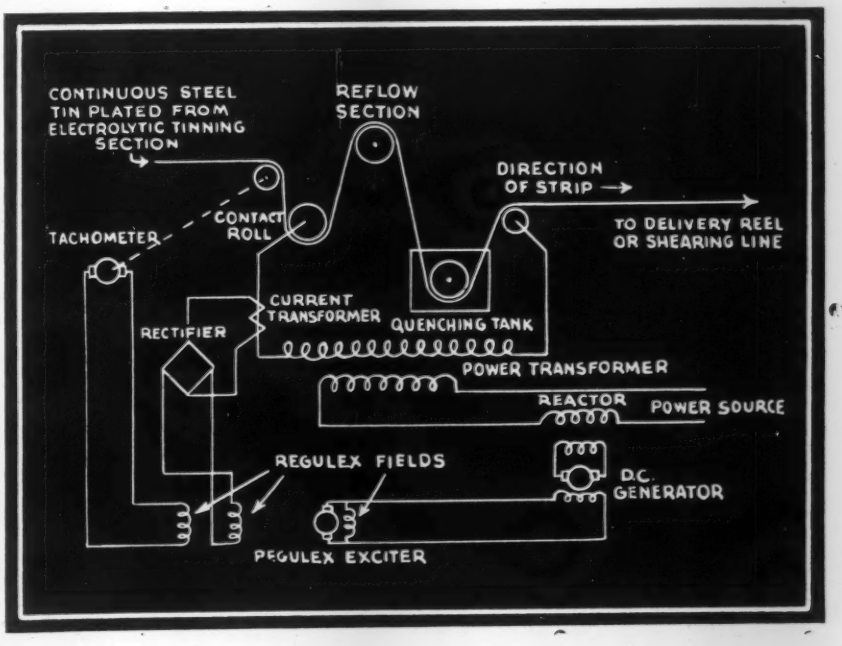
Provides Accurate Control for Reflowing

By A. M. Shong
Control Engineer
Allis-Chalmers Mfg. Co.

ONE of the most vitally needed steel mill products whose production involves extremely accurate control is tin plate. After steel strip is tin-plated in the continuous electrolytic process used, it must be heated just enough to cause the plated tin to reflow in order to give the plated surface a lustrous finish. The problem, therefore, is to impose an accurate control on the reflow current in accordance with speed changes, acceleration and deceleration of the tinning line.

Matches Characteristics of Line

A direct-current machine which can maintain constant an electrically measured quantity, such as current, voltage, torque, tension or speed, the Regulex exciter in this case matches closely the speed and acceleration characteristics of the entire line. Briefly, it is accomplished in this manner:



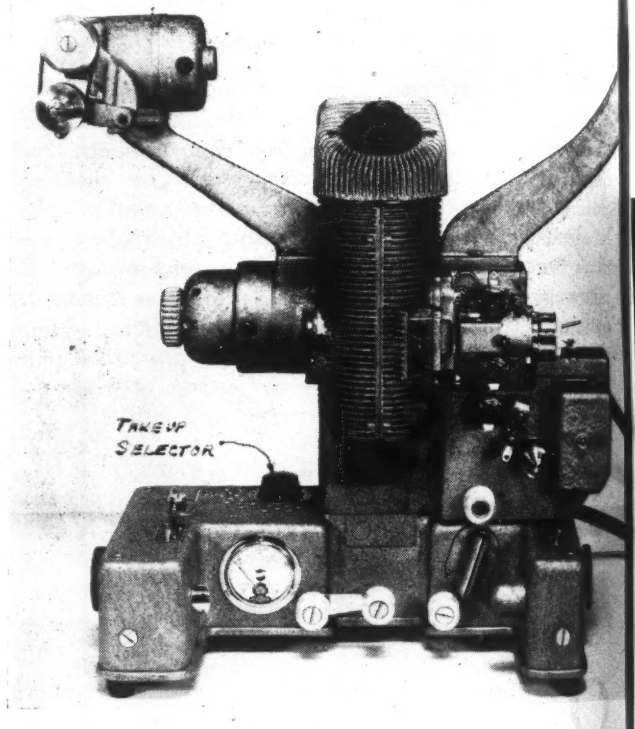
The Regulex machine excites a direct-current generator, which energizes a regulating, saturating reactor. The reactor, in turn, controls the voltage input to the power transformer supplying power to reflow the electrolytically deposited tin.

These voltage changes are lightning-like in their movement, with minimum inductance in a Regulex control field so constructed that at any point on its characteristic curve (exciter characteristics are such that the shunt-field resistance line, saturation curve and air-gap line are made to coincide, for all practical purposes, in one straight line), extremely accurate regulation is provided for these particular control functions.

Thus a more flexible process results from this accurate control, and important increases in the speed of production of tin plate are possible.

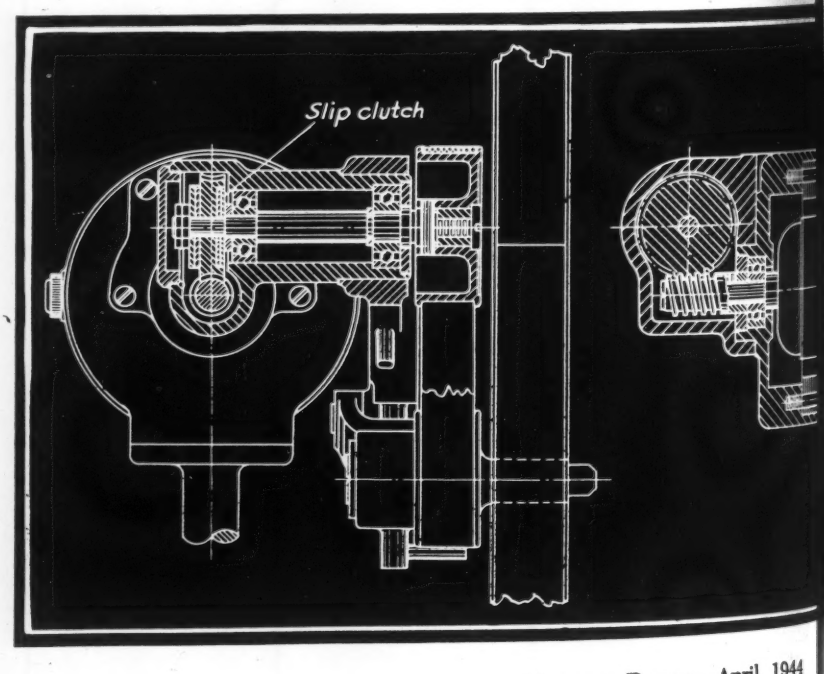
Built-in Slip Clutch Aids Motor Drive

By B. E. Stechbart
Chief Engineer
Bell & Howell Co.



MEETING varying torque and speed requirements, a small reduction-gear electric motor with built-in slip clutch is being used successfully on the projector illustrated to effect take-up and rewinding of 16-mm. motion picture film. The nature of this projector design is such that an independent motor-driven type of take-up mechanism was found to be best suited.

A series-type motor with a specially wound armature and field to meet a certain torque-speed curve characteristic is used. More specifically, the take-up motor must have a low-torque characteristic at high rotational speed and a high-torque characteristic at low rotational speed. This method was used in order to obtain a reasonably uniform film pull, regardless of the reel diameter, which progressively increases as film is being reeled upon it. It was found impossible to obtain a motor torque-speed curve characteristic which would compensate completely for the range of reel diameters encountered, so a rheostat was incorporated in the motor circuit to be used with small reel hubs and when operation is by direct current. A take-up selector knob, shown on the base of the machine in the illustration, operates this rheostat, the different conditions obtainable being clearly indicated on the calibrated dial which is also located in the base.



To rewind film, the motor is reversed and operates at full line voltage to effect a high-speed rewind. In the motor unit, between the driven shaft and the reduction worm gearing, there is a built-in slip clutch, its purpose being to enable the reel on the spindle to be rotated for threading purposes and for removal of slack from the film. This otherwise would be impossible because of the irreversibility of the reduction worm gearing utilized.

In order to provide automatic self-compensation, an additional mechanism has been added to the original motor unit. This addition comprises a take-up

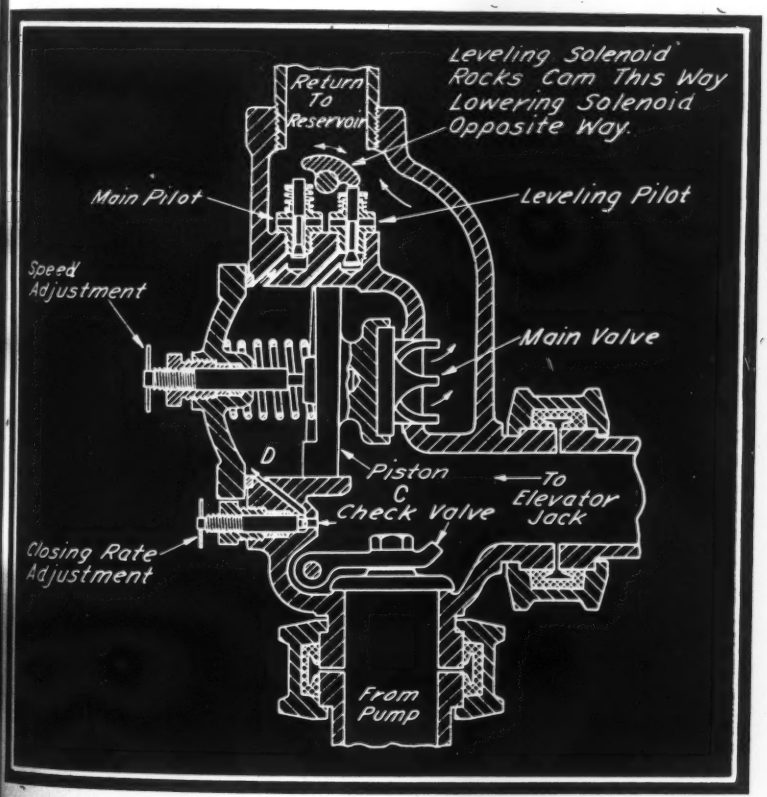
reel spindle which is mounted on a pivoted arm on the take-up bracket and is coupled to the electric motor unit through a flat belt. This flat belt passes over a pulley which is mounted upon the take-up reel spindle bracket and another pulley mounted on the motor unit shaft as shown in the photograph and drawings.

Forward end of the pivoted arm, carrying the take-up reel spindle, is suspended by the flat belt passing over the pulley on the spindle shaft and the arm is spring-urged to keep the flat belt under a slight initial tension. Torque at the take-up reel spindle is governed by the traction of the belt automatically controlled by the weight of the reel and any film upon it.

The flat belt also acts as a slip clutch, enabling the reel on the spindle to be rotated for threading purposes and for removal of slack from the film.

Controls Speed with Helical Cam

By J. B. Harrison
Chief Production Engineer
Rotary Lift Co.



MANY problems are encountered in designing a satisfactory lowering control valve for hydraulic elevators. Of prime importance, however, are smooth starts and smooth and accurate floor stops. The rotary lowering valve shown combines these functions, provides for external adjustment of each and incorporates a check valve in the assembly.

In the illustration shown the valve is in the partially open position, being essentially a pilot-operated, balanced-piston type of conventional hydraulic valve design. The pilot valves are part of the assembly and are opened selectively by operation of the rocker shaft shown through a linkage to two solenoid plungers. The main pilot solenoid is under control of the regular "down" circuit of the elevator, while the leveling pilot solenoid is energized by a switch on the car operated by a cam at each floor to afford automatic leveling. Other designs utilize externally mounted magnetic valves as pilots with the same results.

Differential Pressure Opens Valve

With both pilots closed, line pressure exists in chambers *C* and *D*, holding the main valve tightly seated. Opening of the main pilot causes a pressure drop in chamber *D* and the resulting pressure differential moves the piston to the left and opens the main valve. Closing of the main pilot allows pressure to build up in chamber *D* through an adjustable needle, forcing the piston to the right and closing the main valve. The rate of closing is, of course, dependent upon the adjustment. Rate of opening of the valve (or drop away of the elevator platform) is dependent upon the rate at which pressure is let out of chamber *D* in relation to the bleed-in through the adjustable needle. Provision is made in the linkage for control of this opening rate.

The unique feature of this valve lies in the fact that a second bleed hole has been incorporated part way down the cylinder wall for operation of the valve in a partially closed position. This bleed hole is under control of the second pilot valve called the "leveling pilot". With the main pilot closed and the leveling pilot open, the piston rises only until the upper edge cuts off the outlet through the leveling pilot, at which point the pressures in chambers *C* and *D* assume a balance to hold the

piston stable. The amount the main valve is opened is dependent upon either the location of the leveling bleed hole in the cylinder wall or the length of the piston from valve seat to top edge. The leveling speed could, therefore, be made adjustable by varying either the location of the hole or the length of the piston. In the design shown, the latter is accomplished by making the upper surface of the piston in the shape of a helix or a helical cam, whereby rotating the piston any portion of a revolution varies the effective length of the piston in relation to the bleed hole. This rotation of the piston is accomplished by a speed-adjusting screw, the end of which is flattened to fit into the piston. This screw also serves as an adjustable stop screw to limit the maximum open position of the valve for adjusting the maximum lowering speed of the elevator.

Quick-Acting Brake Requires Minimum Power

By R. M. Scott
Chief Engineer
New England Butt Co.

IN RECENT years the wire cable industry, like nearly all manufacturing industries, has turned to higher operating speeds. Not only have speeds increased, but automatic stop motions, functioning to stop the machine whenever a spool of wire terminates or a wire breaks, have been added to most machines used in the wire industry. These two factors—higher speeds and automatic stops—have made it necessary for better and more powerful brakes to be used on such equipment.

The design of automatically operated brakes for such machines as wire stranders have presented an interesting design problem. Requirements of such a brake are as follows:

1. Quick and powerful stopping. As stranding machines have inherent high rotational inertia, the high speeds require brakes of horsepower capacity several times that of the driving horsepower. Time of stopping is essential as rigid specifications make it necessary to stop the machine before a broken strand has become a part of the finished cable.
2. Means of applying the brakes must be readily adaptable to interlocking with a stop motion type of device (usually electrical), as well as with safety devices to prevent injuries to the operators.
3. Because frequent stopping is a normal condition in the operation of such a machine, low maintenance of brake parts, as well as that of operating parts, is essential.

In the accompanying photograph is such a machine equipped with a brake powerful enough to stop the machine in the required time for efficient operation yet operated by a solenoid which will fit into one's hand. Normal operation requires that the rotating member of this machine, approximately 30-inch diameter, operating at 900 revolutions per minute, be brought to a full stop in not over three seconds. To accomplish this a brake torque of approximately 225 foot-pounds is required for the machine.

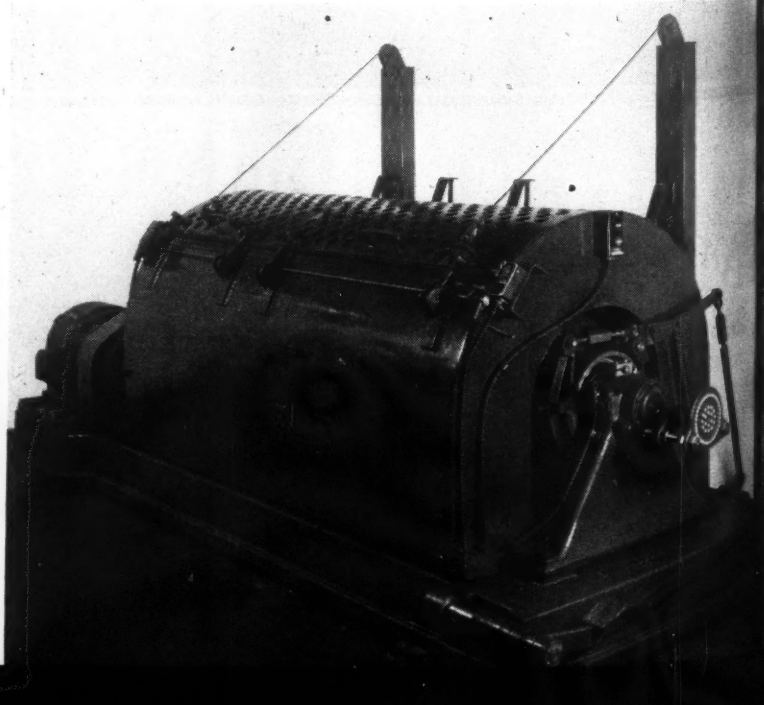
The machine is fitted with a conventional brake at each end of the machine. They are of the externally-contracting, spring-set type using molded linings of about .35 co-efficient of friction.

Various methods of brake control were given consideration. Brake release by means of solenoid was disapproved because a solenoid of sufficient capacity not only would be too large, but also would be too noisy and would soon wear itself out due to the frequent stopping and starting required. A thrustor type brake was not adaptable as the cycle of operation was too slow. Therefore, the design as used was developed and is covered by patents.

Operation of this brake is best illustrated by referring to the accompanying sectional view. It can be seen that the brake is set by a tension spring and is released by an upward motion of a connecting link and arm.

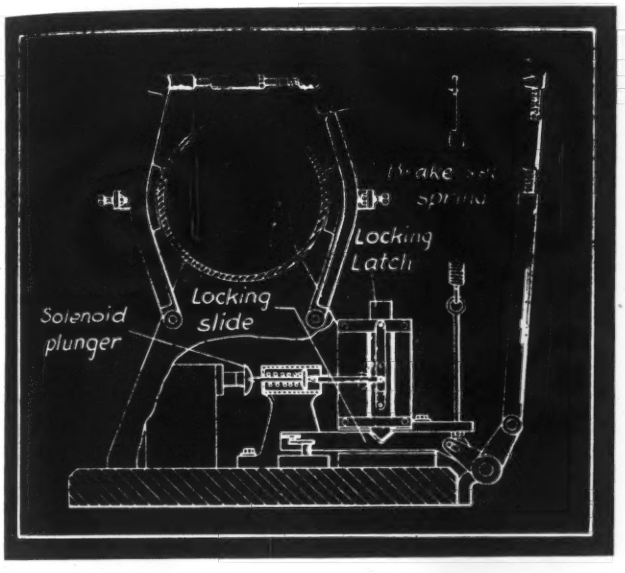
This upward motion is obtained by the operator stepping on one of two foot pedals. This not only releases the brake but also allows the V shaped notch in a locking slide to move back directly under the V-point of a locking latch. This member then drops into the notch, bringing toggle levers into a self-locking vertical position. At the same time a link, with the aid of a spring, pulls a solenoid plunger partially out of its solenoid.

By this operation the brakes are in a released position and locked in that position so that by pushing the start button the machine may be operated. When the motor is stopped, current is applied to the solenoid, thus sucking in the plunger and pulling the toggle links



to the dotted position shown. As the links are pivoted the locking latch is unlocked, thus allowing the brake set spring to apply the brake. By linking several brake assemblies these brakes may be released with one small solenoid, requiring the least amount of electrical energy.

This arrangement requires interlocking in the electrical circuit so that the machine may not be operated unless



the brakes have been set in the running position. The solenoid must also be so interlocked that the solenoid circuit will be open at all times except at the instant of operation directly following the opening of the motor circuit. This is accomplished by use of the following circuits.

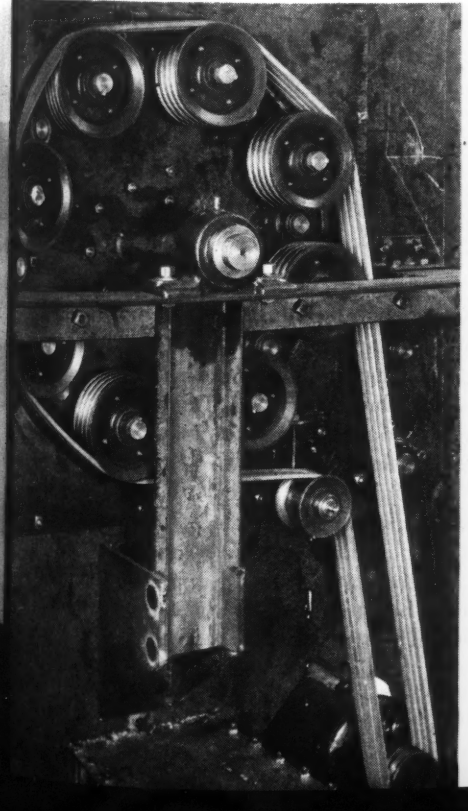
The entire control circuit is operated on low potential, namely, 25 volts, this potential being supplied by a transformer mounted with other controls in a cabinet on the end of the machine.

A lever-operated limit switch is mounted on the base of the machine in such a position that the brake-locking slide will close the contacts when this member is locked in the operating or running position. A low-voltage relay with two normally open contacts and one normally closed contact is used in the starting circuit.

With the toggle switch closed, pressing the "start" button closes two normally open contacts—one of these acting as a holding contact—and the normally closed contact is opened. Then the motor starts, provided the brake has been released, as the starting coil circuit is completed through the limit switch already mentioned. If the brakes have not been released the circuit is left open and the motor will not start. When the main line contactor or motor starter closes, a normally-open contact closes also, but no current will flow through the brake solenoid as the contact on the low-voltage relay is open while the motor is operating.

When any one of the several switches opens in the low-voltage circuit the main line contactor opens, cutting off power from the motor, as the holding coil is de-energized by this operation. However, the normally open contactor which is closed by the motor starter has a delayed action which allows a momentary current to flow through the brake solenoid circuit, thus energizing the solenoid and releasing the brake latch, permitting the springs to set the brake. The opening of this contact then restores the control apparatus to its prior condition ready for the cycle to be repeated.

Due to the fact that current is applied to the solenoid for only a fraction of a second, coil burn-out due to overloads from lack of plunger seating is eliminated even though the brake be applied every few minutes. Large braking forces are thus effectively applied with the use of small amounts of electrical energy, insuring positive quick operation.



Sixteen Shafts from One Motor

By David C. Turnbull
Chief Engineer
American Foundry Equipment Co.

TO POSITION cast camshafts under the blast of two centrifugal blasting wheels, to revolve them in these positions until molding sand is entirely removed, and then to discharge them from the machine constituted the problems associated with the design of the cleaning machine illustrated. These problems were solved by spacing sixteen

shafts in a circle carried by flange type bearings on two circular steel end plates mounted on a central shaft. Each shaft has two small hardened disks spaced longitudinally to match bearing sections on the camshafts.

Camshafts laid on these disks on one side of the machine are carried up to blasting position as the central shaft is rotated by means of a ratchet wheel and pawl arrangement. Action of an eccentric for the ratchet wheel and pawl is adjustable to move the camshafts the distance between the two blasting positions which is equal to the spacing of the carrying shafts. Thus the necessary pause is provided in each blasting position—the time of pause being controlled by a variable-speed drive operating the pawl. Finished camshafts roll out of machine as they move down after being cleaned.

The photograph shows eight 7-inch diameter V-belt sheaves on alternate shafts. This original setup has been changed to one two-groove sheave on each of the sixteen shafts, these being spaced on the shafts to overlap adjacent sheaves. Such an arrangement provides two sets of driving disks for the rotation of the camshaft. Each set of eight sheaves is driven by two V-belts from a motor set on floor level.

Varying from 19 inches to 40 inches long and heavily coated with sand, twenty camshafts are cleaned per minute. The floor space occupied is 5 by 11 feet. An elevator and separator are utilized to remove the molding sand and return the steel shot to the blasting wheels.

Single Drive Coordinates Units

By Fred W. Greer
Chief Engineer
J. W. Greer Co.

A COATING unit for chocolate coating candies, biscuits, crackers, etc., consists essentially of a feeding conveyor, a pre-bottoming attachment which puts a coating on the bottom of the products and a coating machine which completely coats the products. This machine utilizes a supply tank and a pump for circulating chocolate from the tank to a flooding hopper which distributes the chocolate over the products as they pass beneath it on a wire mesh belt. A cooling conveyor, and a packing table conveyor complete the unit.

It is desirable to have the feeding conveyor, pre-bottomer, coating machine, cooling conveyor and packing table conveyor driven from one main motor and variable speed drive. Thus, when it is desired to change the speed of the unit all of the various sections maintain their proper relative speeds without the necessity of individual adjustment.

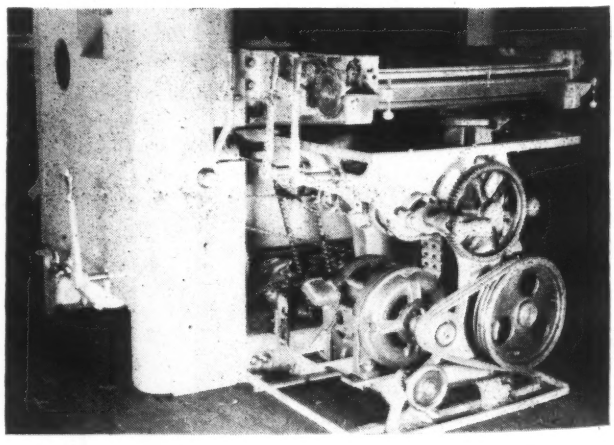
In designing the main motor drive the following specifications had to be met:

1. Compact drive of 3 horsepower
2. Accessibility for cleaning and maintenance
3. Constant low-speed drives for the chocolate pump and tank agitators or paddles
4. Constant high-speed drive for the water-circulating pump for the chocolate supply-tank jacket
5. Variable-speed drive with an individual clutch to the feeding table and pre-bottomer
6. Variable-speed drive for the wire-mesh belt
7. Variable-speed drive with an individual clutch to the cooling conveyor
8. Convenient location and easy operation of clutches and speed control.

Main frame of the drive is comprised of two castings spaced apart with shouldered round steel tie pieces at three corners and with a cast-iron tie piece at the other. This latter serves also as a mounting for the clutch-lever pivot shafts. One of the lower tie pieces is used as a support for the pivoted motor base. A 3-horsepower motor, V-belt drive, and variable-speed unit are utilized together with the necessary gears, sprockets, chains, etc., to provide the drives to the various parts of the machine.

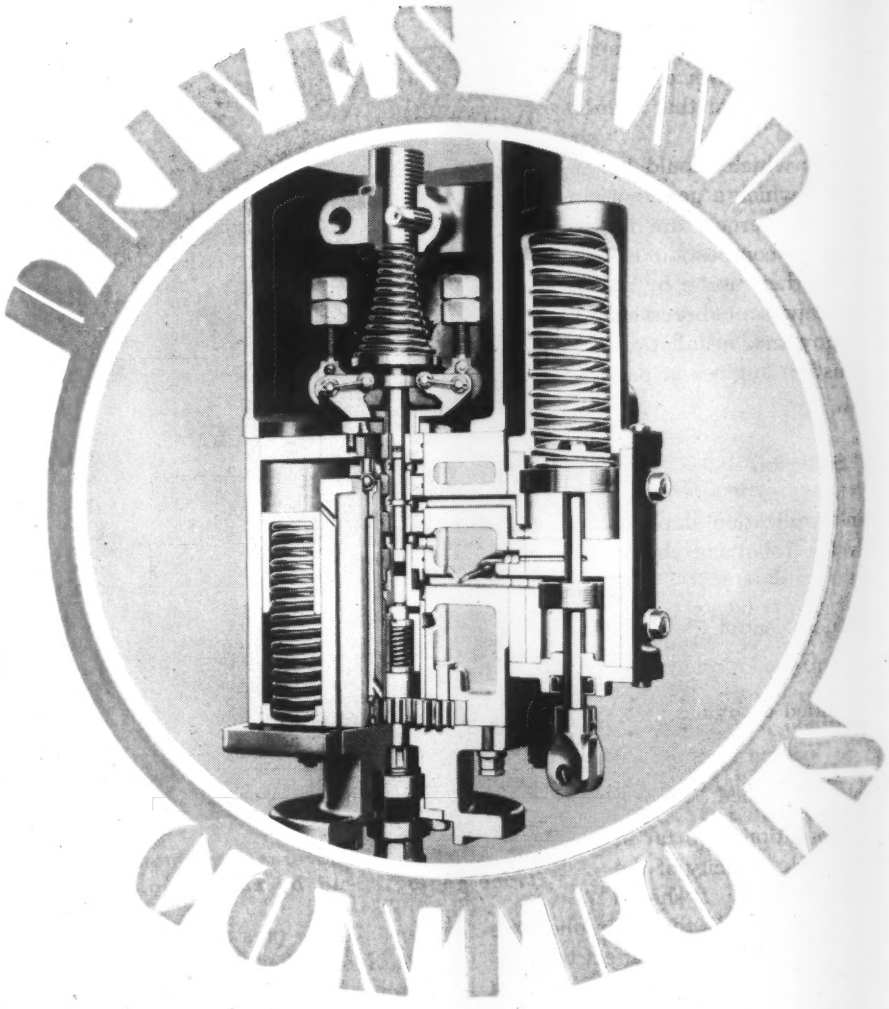
The drive shown in the illustration was designed as a separate unit so that all parts could be completely assembled with the motor, variable-speed drive, clutches, etc., on an assembly bench and then installed below the delivery extension of the coater.

Where gearing is used, the pinions are made of laminated phenolics, thus eliminating gear noises. Chain



speeds are kept low enough to eliminate chain noise. Ball bearings are employed for high-speed shafts and bronze oil-grooved sleeve bearings for low-speed shafts.

All grease fittings are brought to the outside of the drive so that greasing can be done without removing guards. A drip pan is provided below the drive to catch oil, grease, dirt, etc., which may fall from the drive. The vertical, variable-speed unit facilitates maintenance. This compact drive has proved to be efficient, easy to maintain, easy to clean, quiet in operation and free from vibration. It also is economical to build and meets all of the specifications originally drawn up for it.



Basic Factors in Applying Speed Governors

By G. Forrest Drake
Manager of Engineering
Woodward Governor Company

ALTHOUGH automatic regulation of the speed of rotating machines, prime movers in particular, is old, and the operating principles of the simpler governors are fairly obvious even to the uninitiated, governing is a highly specialized field in which the accumulation of know-how from the experience of years is of great importance in the achievement of a satisfactory application. In some degree each job requires individual engineering, ranging in complexity from the design of an adapter to the development of a completely new governor. Because it is not feasible in a few

minutes to impart to anyone sufficient information to make him a governor expert, manufacturers are reluctant to deliver a governor to a customer's specification unless it is known that the application has been carefully engineered.

There are, however, a number of points which should be kept in mind by the designer of a machine to which a governor is to be applied. Although the majority of governors are installed on hydro and steam turbines, internal combustion engines, *Fig.* 1, and airplane propellers, *Fig.* 2, their use is by no means restricted to these applications. The present discussion is concerned with those points pertinent to governor installations generally and governors for internal combustion engines in particular.

Factors Affecting Governor Selection

Selection of a governor for a given application depends upon a number of factors, among which the following should be carefully considered:

1. Force required to move the speed (or fuel) control
2. Speed range desired
3. Whether isochronous operation is wanted
4. What overspeed is acceptable under specified conditions
5. Engine characteristics
6. Nature of the load
7. What auxiliary or nonstandard controls are needed.

The foregoing points deserve some elaboration. So far as the first is concerned, it is obvious that the work capacity of the governor must be ample to handle the necessary load. It should be noted that the necessary work which a governor of the hydraulic type has to do does not, as in the case of the mechanical governor, affect the governor sensitivity. It does, however, affect the servo-speed for a given pilot valve opening; the greater the load, the higher the back pressure and the lower the pressure drop across, and therefore the flow through, the pilot valve. The result is greater speed deviations upon sudden load changes than would be experienced if the governor were lightly loaded. It is desirable that the actual load not exceed 30 or 40 per cent of the maximum work capacity of the governor; for instance a unit rated 8 foot-pounds should not be required to do over 2.5 to 3 foot-pounds of work per stroke for best results.

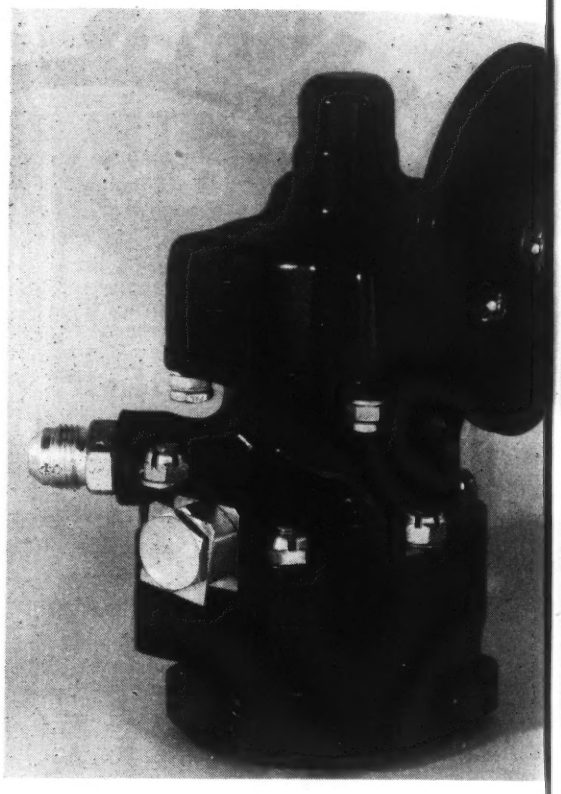
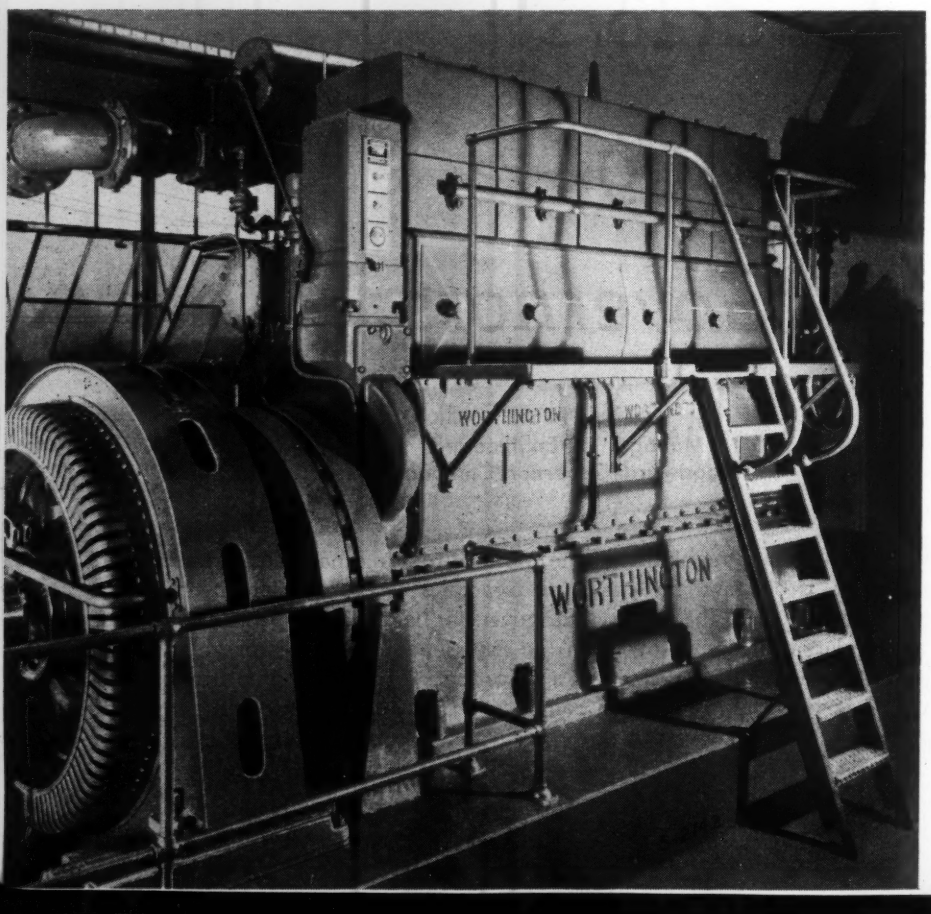


Fig 2—Propeller governors maintain constant airplane engine speed by controlling propeller pitch

Fig. 1—Below—Governor should be mounted close to engine flywheel and provided with as direct a drive as possible



Speed Range and Performance

The narrower the speed range required the more precisely can the unit be designed for best performance. For example, the pump capacity required for the desired maximum servo-speed at low governor speed might result in overheating if run continuously at four times that speed; the ballhead stability and the compensation characteristics required for optimum performance vary with the speed, and for wide speed ranges must be compromised. In other words, a unit intended for a single operating speed can more satisfactorily be fitted to the conditions obtaining at that speed than can a unit which must cover a four-to-one speed range, although governors giving

satisfactory performance for special applications have been built to cover ranges as high as six to one.

Some applications may require isochronous operation (constant equilibrium speed regardless of load). The maintenance of 60-cycle alternating current supply is one such. However, it should be remembered that if two or more units are to be paralleled, only one can be isochronous,

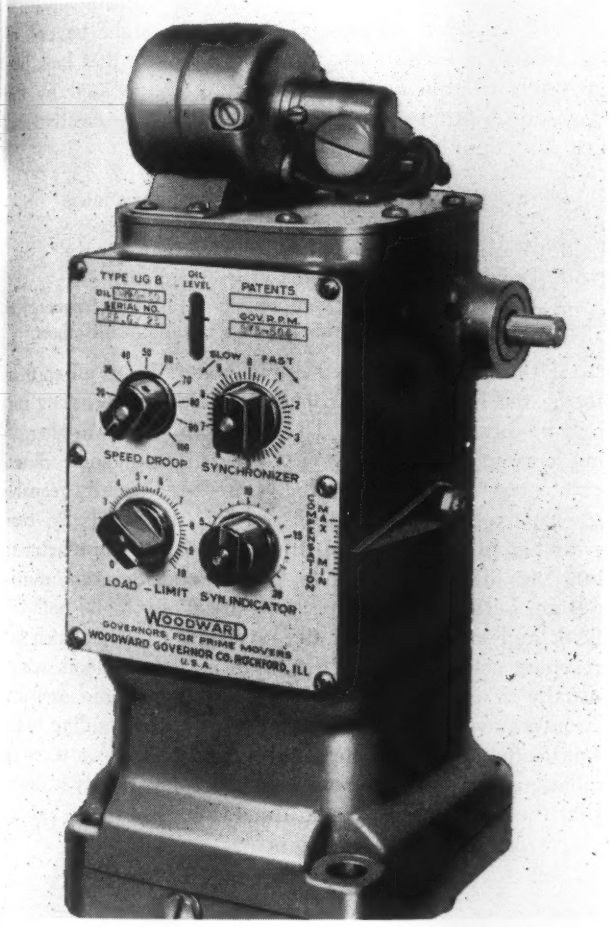


Fig. 3—Synchronizing motor on top of governor permits varying speed when generator unit is being paralleled

and that one must take all sudden load changes. The other units must be of the speed-droop type (decreasing speed with increasing load) in order properly to divide load. Readjustment of the latter as the isochronously governed unit reaches maximum or minimum load is necessary in such an installation.

In some instances the overspeed allowable when all or some designated fraction of the load is dropped is specified. This depends as much on engine characteristics as on those of the governor. To predict this it is necessary to know the type of prime mover, the WR^2 of the rotating system and the magnitude of the load change. Large WR^2 helps to keep down swings on load change under normal conditions, but may cause resonance troubles as discussed later.

The nature of the load is important, since some loads (such as marine propellers) are more stable than others. Also some loads, as the example mentioned, are steady, while others such as power supply to a rolling mill are subject to sudden changes of large magnitude. Obviously the control problems presented differ.

Many auxiliary devices are available, such as automatic synchronizing equipment, means for remotely adjusting speed, *Fig.* 3, load-limiting devices, *Fig.* 4, shutdown devices operated by air, oil or electricity, and many others. Most such controls now available were designed to meet the requirements of some specific application. Others will undoubtedly result from future problems.

Provide for Governor in Early Stages

Satisfactory performance of a governor, once selected, depends upon a great many things for which responsibility frequently falls unjustly upon the governor manufacturer. Some of these are due to a tendency (which fortunately is diminishing) on the part of the machine designer to provide for the governor as a sort of afterthought. The result has been that, on occasion, it has been necessary to mount governors surrounded by exhaust pipes or otherwise made inaccessible, driven from the springy end of a chain-driven camshaft, and required to operate a fuel control of badly nonlinear characteristics through a complex linkage full of lost motion and friction. Although it is hardly feasible to determine first the governor drive, fuel control and linkage, and design an engine around them, the governor is so important to the satisfactory performance of the engine that careful consideration, *during the design process,* of the desirable conditions for best governing results would save much money and many harsh words. These desirable conditions, in the case of internal combustion engines, may be outlined as follows:

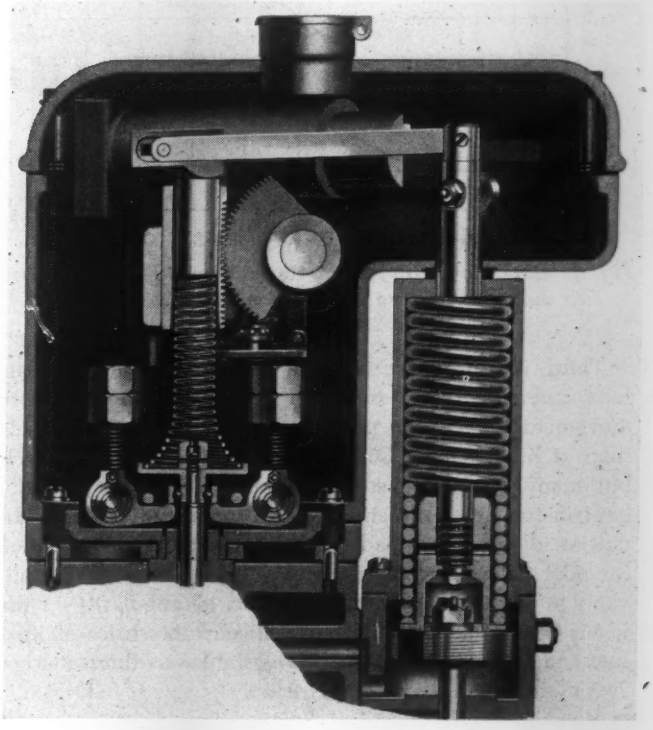


Fig. 4—Travel of power piston on torque-limiting governor is controlled by torque-limit cam engaging roller

First, since to meet regulation demands the governor must be sensitive (response to .01 per cent speed change is common) and fast in action (full stroke in 1/10-second in some models), the speed of rotation of the ballhead should be truly proportional to that of the prime mover. This suggests that the drive should *not* be through any shaft subject to periodic variations in torsional deflection (such as a camshaft), nor by a floppy chain, nor by poorly cut gears, otherwise the governor will respond to the resultant speed changes by undesirable jiggling of the fuel control. The ideal drive would be direct from the flywheel end of the crankshaft, *Fig.* 1. This is seldom feasible, but every effort should be made to keep from the governor drive all speed fluctuations which are not representative of engine speed changes. Satisfactory governing on engines in which the drive was far from the ideal has been provided by the employment of such devices as spring drives, special porting and the like, but these are definitely to be avoided if possible.

Second, the governor should be accessible. While a reputable governor is at least as troublefree as most mechanical devices, it is still desirable to be able to change the governor or its adjustments without removing the exhaust manifold, number one cylinder, or the crankshaft. This consideration is one of convenience only, unless the location is such as to produce excessive ambient temperatures, as in the case of the exhaust pipe barricade, in which case performance is adversely affected.

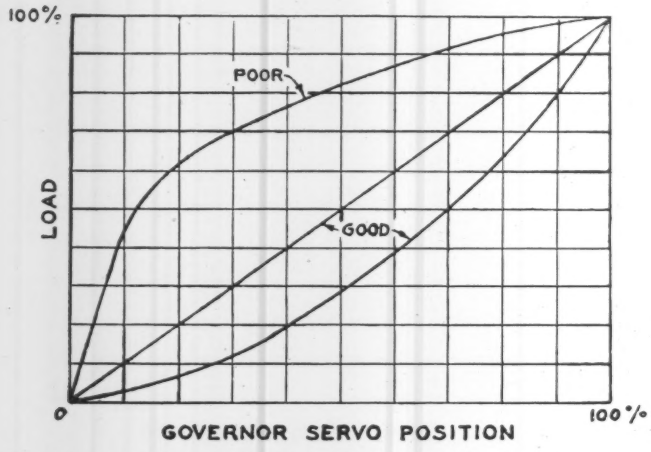


Fig. 5—Good governing characteristics at low load require flat curve of fuel valve position in relation to load

Third, it is desirable that the fuel valve position versus load curve be linear, or better still that it require more movement for a given per cent load change at low loads than at high loads. Diesels are not bad in this respect, but many gas and gasoline engines have a curve badly bent in the other direction, which makes governing difficult at idle and low-load conditions. Some compensation for poor valve characteristics may be made by taking advantage of the foreshortening effects of angularity in the levers of the linkage. The overall characteristics of linkage and valve should give curves similar to those marked "good" in *Fig.* 5.

Fourth, the fuel control linkage should be as simple as possible, operate easily and be entirely free of lost motion, unless provision is made always to have the load in one direction. This method of compensating for many joints or sloppy machine work is simple, but it has the obvious disadvantage of requiring greater governor capacity. Lost motion in the fuel control linkage is fatal to stable control.

Even with all the foregoing satisfactorily provided for and the proper governor perfectly adjusted, the results may look like governor trouble. Of course it may be, but before reaching for the telephone to call for governor service, consider the possibility that it might be one of the following conditions, which give results that look like governor trouble and are frequently aggravated by the governor in proportion to the degree that it is sensitive and fast in action:

1. Regularly recurring weak or missing cylinders
2. Fluctuation in fuel supply, particularly gas
3. Regular fluctuations in load
4. Improperly functioning voltage regulator
5. Resonance of firing frequency with natural frequency of the machine either alone or in parallel with others.

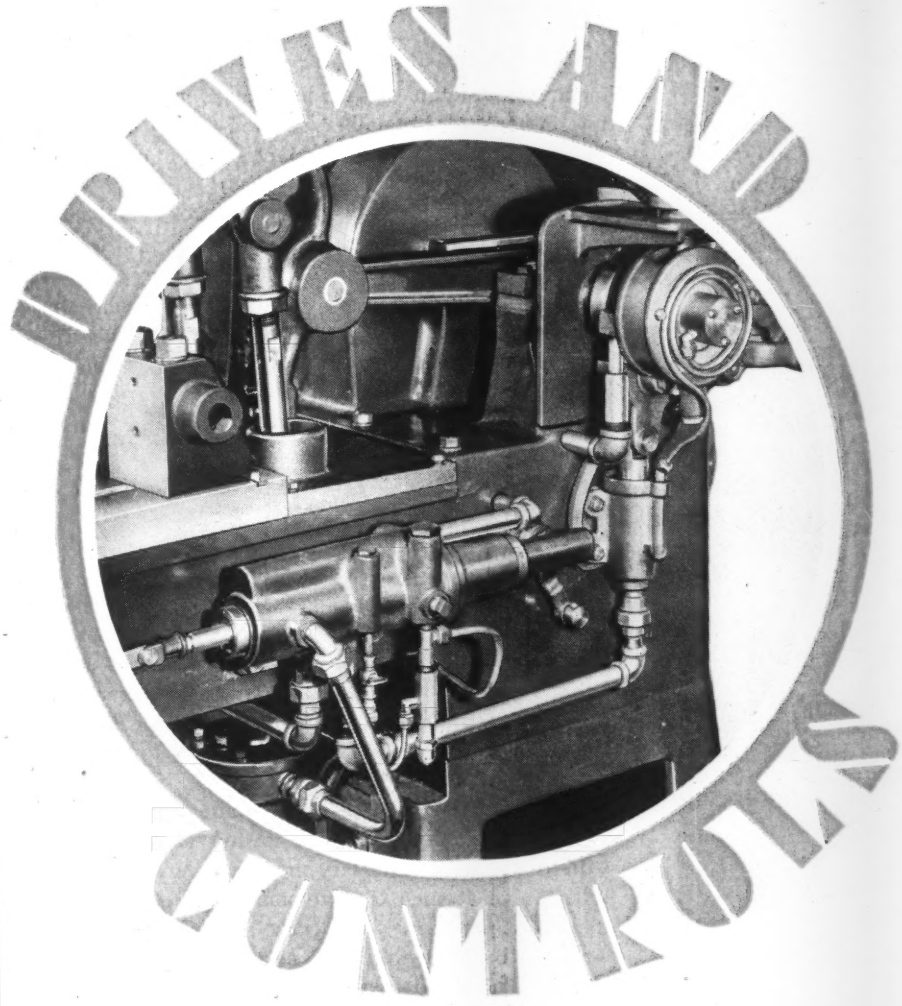
Of these the first three apply generally to all sorts of loads; the fourth to electrical only, either alternating current or direct current; and the last principally to alternating current, although it might occur with some direct-connected loads. The first four are usually capable of relatively simple correction in the field by competent representatives of the equipment manufacturers, but the fifth must receive proper consideration during design, because field corrections are likely to be both difficult and expensive. In the electrical case in which this resonant condition exists the result is periodic load swings, usually at camshaft frequency, as the machine oscillates about an average speed which is that corresponding to the line frequency. Of course the governor follows these oscillations and the erroneous conclusion frequently is drawn that the governor is causing the swings.

Where Engines Operate in Parallel

The foregoing implies connection to a large system approximating an infinite bus. When the system consists of two or more engines in parallel, there are several resonant frequencies which should be kept as far as practicable from possible forcing frequencies, especially if the paralleled units are greatly different in characteristics. In either case, if forcing and natural frequencies are close and the forcing impulse severe, the oscillations may reach such violence that the machines pull out of synchronism and are impossible to keep on the line.

This matter is discussed at some length because, while the addition of WR^2 to an engine usually improves governing results, care should be taken to be sure that such addition does not result in the resonant condition described.

Other fields of application may suggest themselves, for example speed control of electric motors or continuously variable transmissions (such as the hydraulic pump and motor combination provides. In these as in other applications the desirability of consulting the governor manufacturer early in the design cannot be overemphasized. Use of his special knowledge may make the difference between a successful or an unsatisfactory job.



Fundamentals of Hydraulic Valve Selection

By William H. Stanton

SELECTION and application of hydraulic valves can be undertaken on a competent basis only when there exists a clear understanding of their functions. Probably these can best be demonstrated by a brief analysis of the basic elements used in all hydraulic systems—the reservoir, the pump and the conversion unit.

Reservoirs for the fluid generally are enclosed and designed so as to withstand safely the top pressure produced by the pump, plus the static fluid head and a reasonable factor of safety. Pumps may be any one of three predominant types: Gear,

vane or piston. Producing pressurized flow, they are what might be termed the "muscle" of hydraulics.

Finally, there are the conversion units, the two types used in most applications being the cylinder and piston, for conversion of hydraulic energy into linear motion, and the hydraulic motor, for conversion into rotary motion.

If the development of modern hydraulic systems required the utilization of only these three basic units, the machine designer's job in applying hydraulic power and control would indeed be simple. However, while this three-unit combination competently produces power, the power is *uncontrolled*. Therefore, before the system can be considered complete, control—by means of valves—must be added.

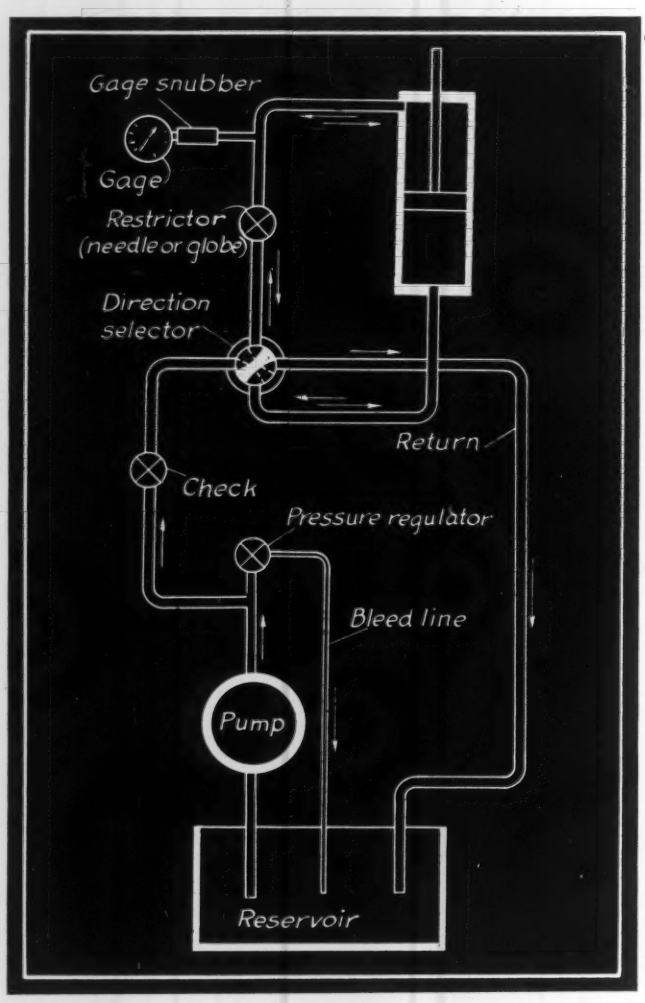


Fig. 1—Schematic of a basic hydraulic system shows how valves are used to control pressure, direction and volume

Hydraulic valves are available in many types and these types take on a variety of forms. However, in all cases it is the *kind* of control required which determines the type of valve to be used. There are four basic kinds of control utilized, these being:

1. Pressure
2. Volume
3. Direction
4. Time.

Just why these kinds of control are necessary might best be illustrated by reference to a basic hydraulic circuit such as that shown in *Fig.* 1. In this circuit, the rate of linear mechanical motion of the piston, produced by forcing fluid into either of the chambers located at each end of the cylinder, is dependent on four factors: Fluid pressure, fluid volume, effective area of the piston face, and magnitude of the opposing load.

Assuming that the area of the piston face is fixed and the opposing load is known, precise control of the rate at which the piston moves will depend, of course, on accurate regulation of fluid pressure and volume. Thus, a pressure-control valve and a restrictor or volume-control valve are inserted between the pump and the cylinder. In the circuit under discussion (*Fig.* 1), pressure regulation can be effected readily through use of a relief valve such as is shown in *Fig.* 4*c*.

Location of Regulator Important

To permit the required control of pressure without imposing restriction on the flow, the valve is not inserted directly into the pressure line, but is mounted just off the line. It readily will be seen that, as the pump builds up pressure throughout the line feeding the cylinder, all of the units in the line will be subjected to approximately the same amount of pressure. Thus, if the pump develops, say, 500 pounds per square inch pressure, and it is desired to operate the piston at 450 pounds per square inch, it is necessary merely to set the spring in the relief valve at such compression that the ball will lift off its seat at piston pressure (plus the small amount of pressure lost between the valve and cylinder due to line friction). The oil by-passed by the valve is directed back to the reservoir through a bleed line.

As installed in the circuit shown the valve fills a double purpose. It not only regulates the pressure in the system, but serves also as a safety vent for the pump. In other words, should the pressure line become blocked for any reason, the pump will be able to relieve itself through the valve. Without the valve, the pump, after building up its maximum pressure, would impose a rapidly increasing overload on its motor.

Thus the fluid is made available to the cylinder at a pressure which is controlled within close limits. But, as has been indicated, the rate at which the piston moves is also dependent on the amount of fluid passing through the line in a given unit of time. Obviously this amount cannot be increased and, if regulation is to be effected, can only be decreased. This decrease is accomplished by some kind of restricting valve which accurately meters the rate of flow. Needle and globe valves, representative types of which are illustrated in *Fig.* 5*a* and 5*b* respectively, are used commonly for this purpose.

There is another type of valve which works on the restriction principle. It is the gage snubber shown mounted just below the gage. Its function is to protect the delicate gage mechanism against sudden surges of pressure which might be developed by pump pulsations or "fluid hammer" effects. All snubbers or so-called "gage savers" function on the same principle, i.e., drastic restriction of rate of flow. In other words, while normal rates of pressure change are afforded ample time to be registered accurately on the gage, sudden surges are subjected to a

shock-absorbing "snubber" action.

Thus far, the circuit in *Fig.* 1 has been developed into a system which includes a reservoir, a pump, a pressure regulator, a flow restrictor, a gage snubber, and a piston and cylinder conversion unit. As such, fluid from the single pump could be used to move the piston in only one direction. Since it is necessary to move the piston in the opposite direction also, some means must be employed to guide the flow first to one end of the cylinder and then to the other. This is done by utilizing a direction-control or selector valve. The type shown in the circuit diagram is a four-port cylindrical plug valve.

It will be seen that the rotor is designed so as to permit flow first to one end of the cylinder and then, upon being turned through 90 degrees, to direct the flow to the other end. In both cases the return to the reservoir is effected through the same port. In addition to the rotary type shown, piston types are available and may be obtained for foot, hand, cam, pilot or solenoid operation.

Applying the Check Valve

To prevent the piston from suddenly releasing its load when the pump is stopped, it is necessary to insert a check valve in the line between the selector valve and pump. Thus, in the present instance, while flow is permitted from pump to cylinder, any attempt at reversal back through the idle pump to reservoir will be checked.

Check valve location can have a profound effect on the functioning of a system. For example, in this particular circuit, it would not be practicable to insert a check valve between the selector valve and the cylinder as this would block the system and make raising of the piston impossible.

Almost numberless hydraulic circuits, more intricate than that shown schematically in *Fig.* 1, have been developed to fulfill the requirements of special applications. One such circuit, shown diagrammatically in *Fig.* 2, represents the hydraulic components of a recently developed cable-stretching machine. This machine is required to prestretch cables which are used for the operation of controls in aircraft. To make possible the testing of five different cable sizes, it was necessary to design the unit so that it would develop accurate pressures to

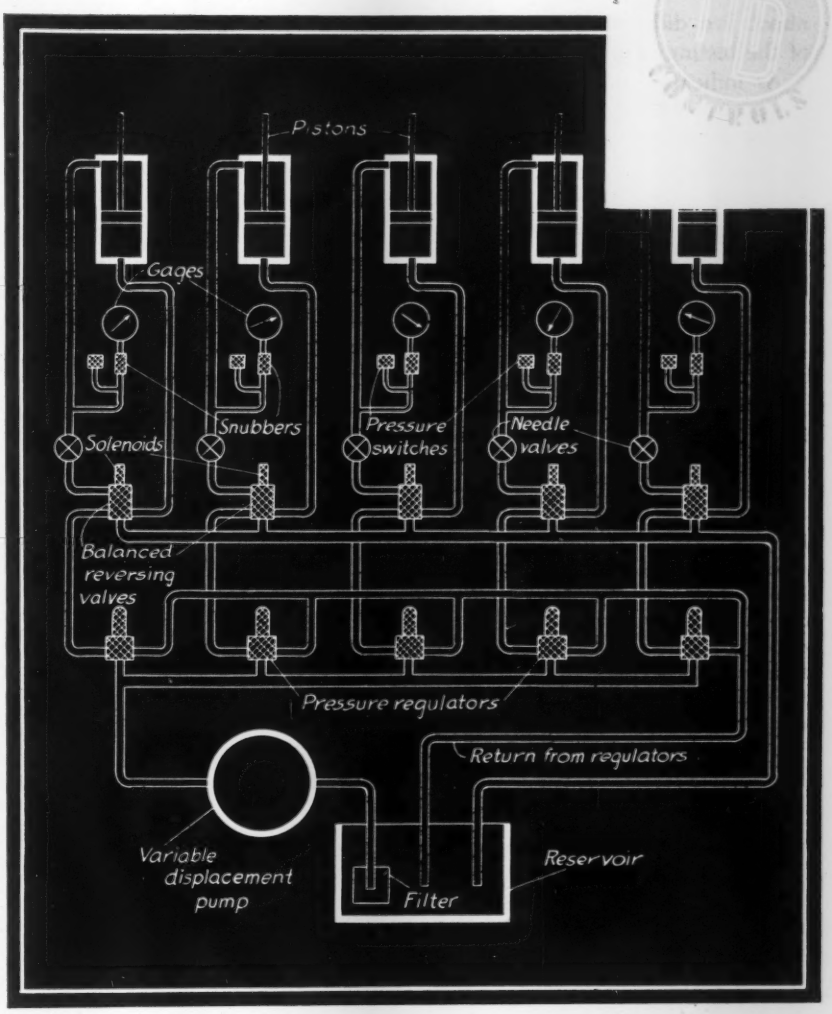
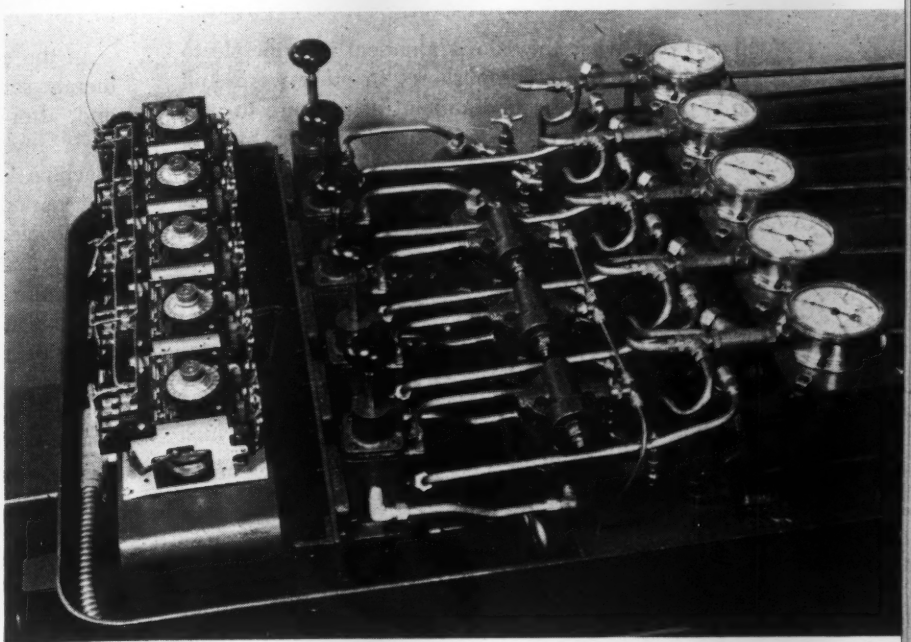


Fig. 2—Above—How valves are utilized to supply controlled hydraulic power to multiple conversion units, each of which requires a different pressure

Fig. 3—Below—Top view of machine for pre-stretching aircraft control cable. Hydraulic system employed is substantially that shown schematically above



effect five different tensions. In addition, precise timing of the testing operations was necessary.

As indicated in the diagram, filtered oil is drawn from the reservoir by a single pump, the output of which is sufficient to supply pressure to five separate piston-and-cylinder units at one time. Upon leaving the pump, the first kind of control to which the oil is subjected is pressure regulation. This is accomplished by a series of five adjustable regulator valves connected to a common manifold. These five valves are set so that each permits a different pressure to pass through it. Thus, the five pistons can be operated independently of each other or in concert.

While the mechanism inside these regulators differs from that used in the simple relief valve, it should be pointed out that what they do is identical in principle to what the simple valve accomplishes. In other words, all they do is permit a prescribed amount of pressure to pass through them, their overload flow being by-passed back to the reservoir.

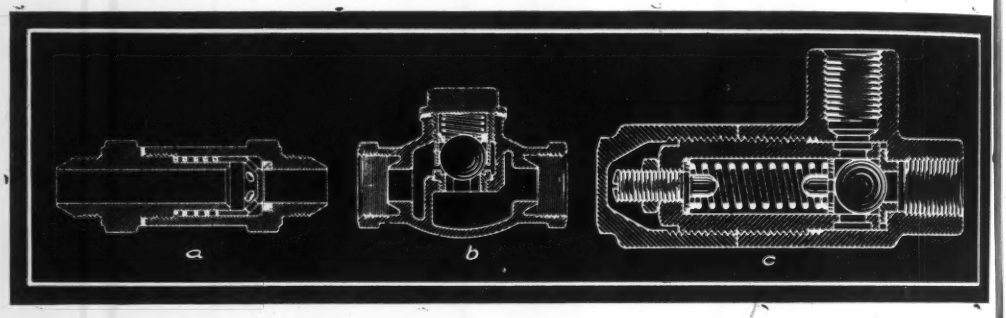


Fig. 4—Right—Representative check and relief valves. Drawings a and b show straight cone-type and globe-type checks respectively, while c illustrates a simple form of ball-type relief valve

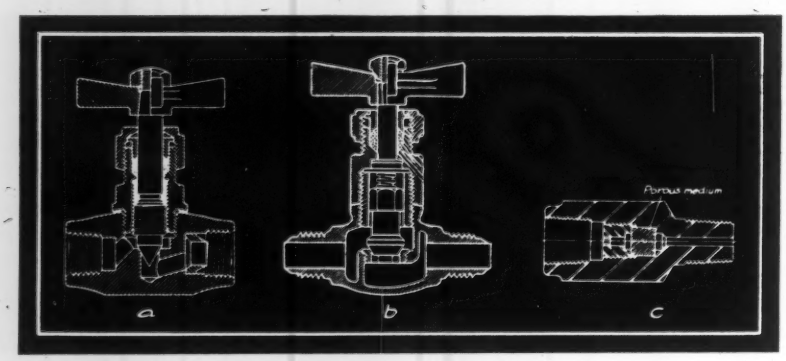


Fig. 5—Left—Simple restrictor valves. Shown at a and b, respectively, are straight needle and globe types. Gage snubber is shown at c

After leaving the pressure regulators, the oil is valved through five separate solenoid-operated reversing or selector valves. Here again the function is identical to that effected in the basic circuit previously discussed. All that these reversing valves do is direct the oil flow first to one end of the cylinders they feed and then to the other.

During the power half of the cycle (pistons moving in such direction as to stretch cables), the oil, after leaving the reversing valves, is fed through needle valves which by regulating the rate of flow into the cylinders, serve to control the rate at which the pistons move. In addition, these needle valves dampen line surges which, if permitted to pass unhampered into the cylinders, might subject the cables being tested to abrupt impact stresses.

It will be noted that the pressure side of each cylinder intake line is provided with a gage and gage snubber. The gages, directly calibrated in pounds tension on the cables, indicate to the operator whether or not the units are functioning at their prescribed pressures, and, as was the case in the basic circuit, the snubbers protect the gage mechanisms from possible damage due to abrupt pressure surges.

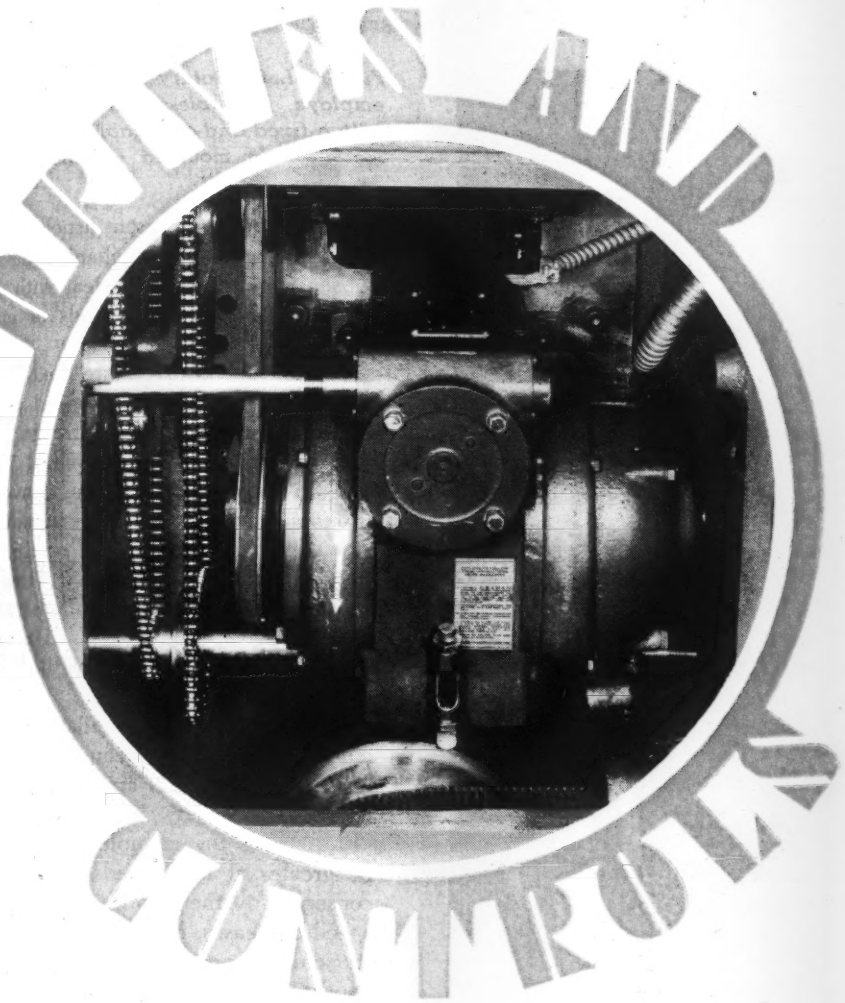
The length of time for stretching the cables is controlled through a combination of hydraulically actuated pressure switches—located in the gage branch circuits—electrical timers, and the solenoids which energize the reversing valve controls. These pressure switches are adjustable and, when a given cable is at the proper predetermined tension, the switch completes a circuit to a timer. After the prescribed time interval for stretching has elapsed, the timer operates a solenoid which pulls its reversing valve to the release position.

This machine is an outstanding example of how hydraulics can be utilized to achieve dependable automatic operation. It is representative of the trend toward simple, foolproof control which, because it permits high efficiency from unskilled operating personnel, is a vital factor in the achievement of low-cost quantity production.

In the case of this machine, the operator is required to merely set an electric timer, adjust a pressure regulator and, after securing the cable ends, release the hand lever of a reversing valve. Without further attention, the machine automatically measures the test pull and the test period, releases the tension and resets itself for the next cycle.

There are, of course, many types of valves employed in hydraulic systems other than and, in many cases, in addition to those herein discussed. However, a clear grasp of the fundamental principles underlying the selection and application of these simple types should go far toward establishing a better overall understanding of that phase of hydraulics which is so extremely vital—control.

The author is pleased to acknowledge the collaboration of the Racine Tool and Machine Co. in supplying the photograph heading this article which shows the arrangement of key units in a hydraulic system used on a metal cutting machine. *Figs. 4a, b, c; 5a, b;* and *Fig.* 3 are through courtesy of The Parker Appliance Co. *Fig.* 5c appears by courtesy of Logansport Machine Inc.



Specifying Variable-Speed Drives

Part I—Mechanical

By Colin Carmichael

COUNTLESS applications for which infinitely-variable speed drives are either essential or highly desirable have inspired inventors to unusual activity in this particular field. Of the numerous ideas advanced many have been discarded as impractical but enough have withstood the tests of hard service to confront the designer with an interesting choice of methods from which to select a drive. In the series of articles of which this is the first, basic considerations in selecting transmission units, and the special features of the major types available, will be discussed. In dealing with

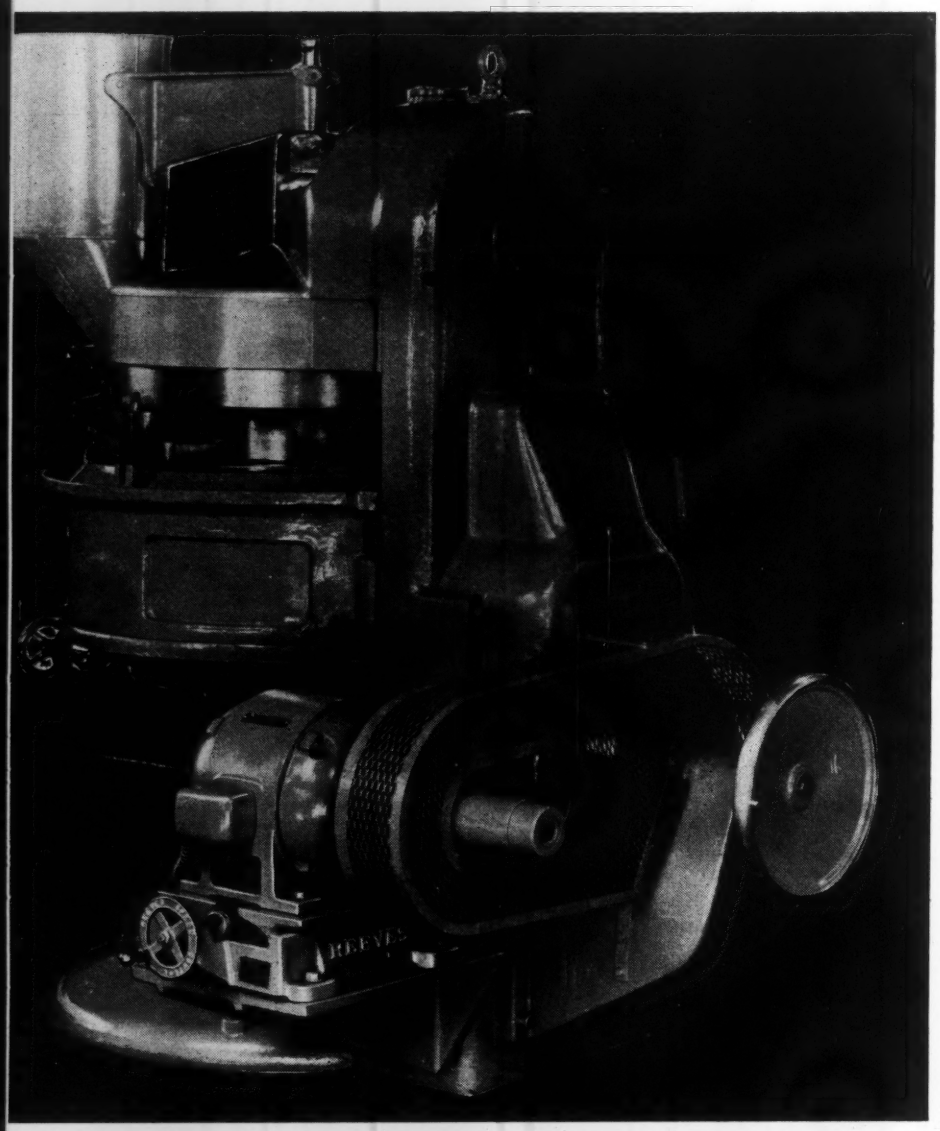


Fig. 1—Left—Rotary pelleting machine employs variable-speed transmission with a fixed and a variable-pitch pulley, motor being mounted on sliding base

specific units, the present article will be concerned only with mechanical devices. Subsequent articles will deal with electrical and hydraulic methods of effecting speed changes.

Variable-speed drives are used for one or both of two purposes—speed change and torque conversion. Although in the majority of cases speed control is the primary requirement, torque conversion also is desirable in order to maintain uniform efficiency, and most transmission units include this feature. How the torque and horsepower requirements will vary as the speed is changed depends on the characteristics of the driven equipment.

On most types of installations torque requirement is fairly constant and the power transmitted is approximately proportional to speed. In other cases, conveyor drives for example, the load may be greater at low speeds than at high, consequently the torque increases as the speed decreases, requiring something approaching constant horsepower. Still another characteristic is found in fans and blowers, where torque decreases as the speed drops.

Where the range of output speeds is extremely high, provision of constant horsepower over the entire range is apt to involve prohibitively large units because of the high torque at low speeds, hence it is important to avoid specifying constant horsepower unless the characteristics of the driven equipment absolutely require it. Transmissions which can go down to zero output speed (infinite speed ratio) obviously cannot furnish constant horsepower as this would involve the absurdity of infinite torque.

For the great majority of machine applications, mechanical transmissions are eminently suited. Where the range of output speeds needed is not too large, V-belts running on variable-pitch sheaves or pulleys are widely used.

If the ratio of output speeds is limited to not more than about 4 to 1, one of the pulleys may be of fixed pitch while the other is variable, the driving motor being mounted on a sliding base as in *Fig.* 1. Speed control is effected by turning a handwheel which adjusts

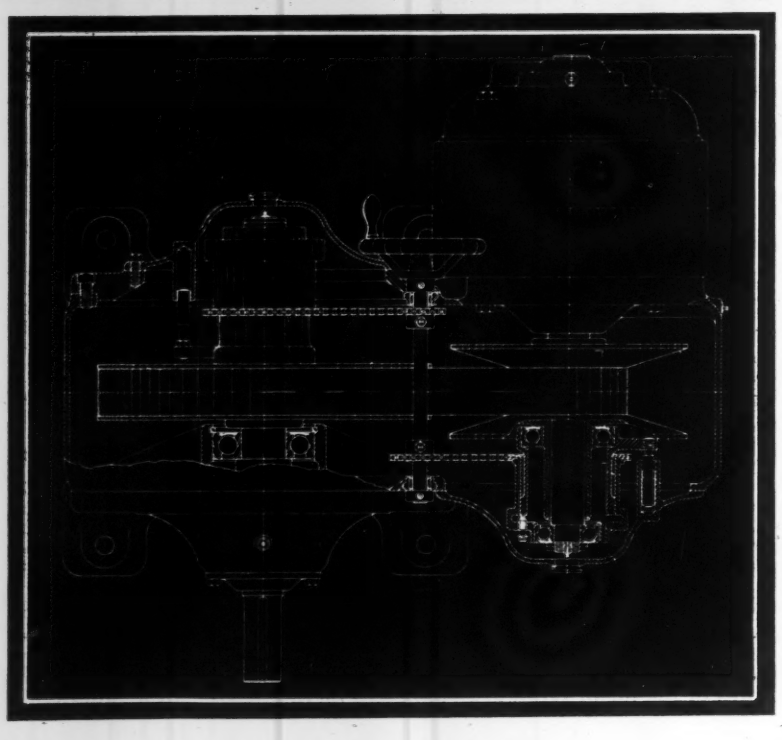


Fig. 2—Left—Cross section of a transmission unit having two variable-pitch pulleys, both positively controlled

the center distance by sliding the motor toward or away from the driven shaft. The two facing cone-shaped disks which constitute the variable-pitch motor pulley are pressed together by a spring which permits the cones to adjust themselves to the correct pitch yet maintains adequate belt tension. Application of this type of drive to the rotary pelleting machine shown in *Fig.* 1 permits the production of pellets for ammunition in a wide range of sizes.

Wider speed ranges are obtained by varying the pitch of both pulleys, in which case the center distance is fixed. Speed is adjusted either by varying the pitch of one pulley,

Fig. 3—Shows bare transmission and separate motor integrally built into a machine housing

using a spring to control the other, or by positively varying both pulleys. Two examples of the latter type of transmission are shown in *Figs.* 2 and 3. It is claimed that positive variation of both pulleys permits more accurate speed control, inasmuch as a spring-loaded pulley may change pitch slightly as the load changes, due to increased belt tension and lateral pressure between belt and pulley groove.

When the transmission unit is built into a machine it is not always necessary to use a completely enclosed unit, and available space often can be utilized most effectively by em-

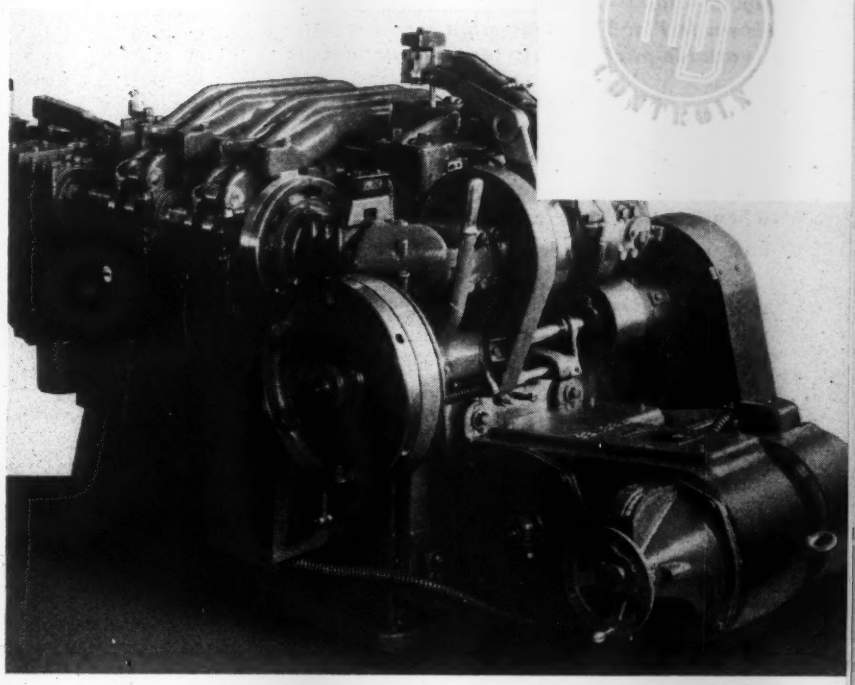
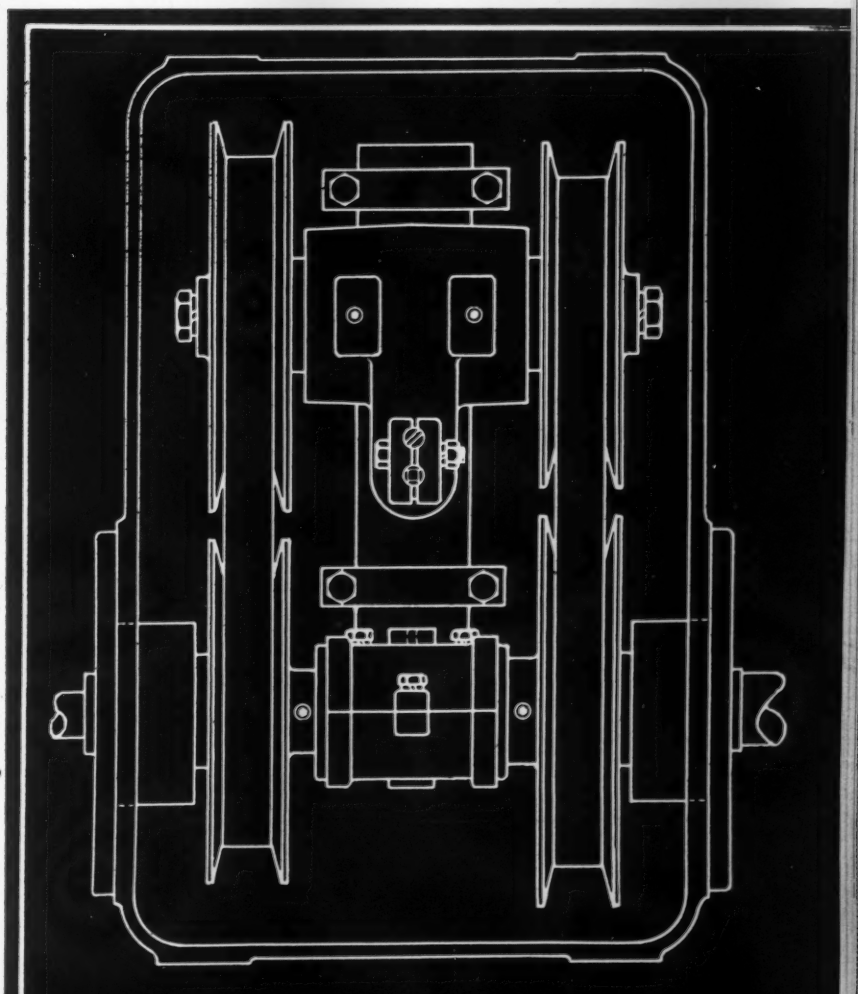


Fig. 4—Above—For outside mounting, enclosed unit with integral motor often offers neat solution, as in this multi-slide forming press

Fig. 5—Below—Compounding provides large ratios without extreme changes in pulley diameters, permits lining up driving, driven shafts



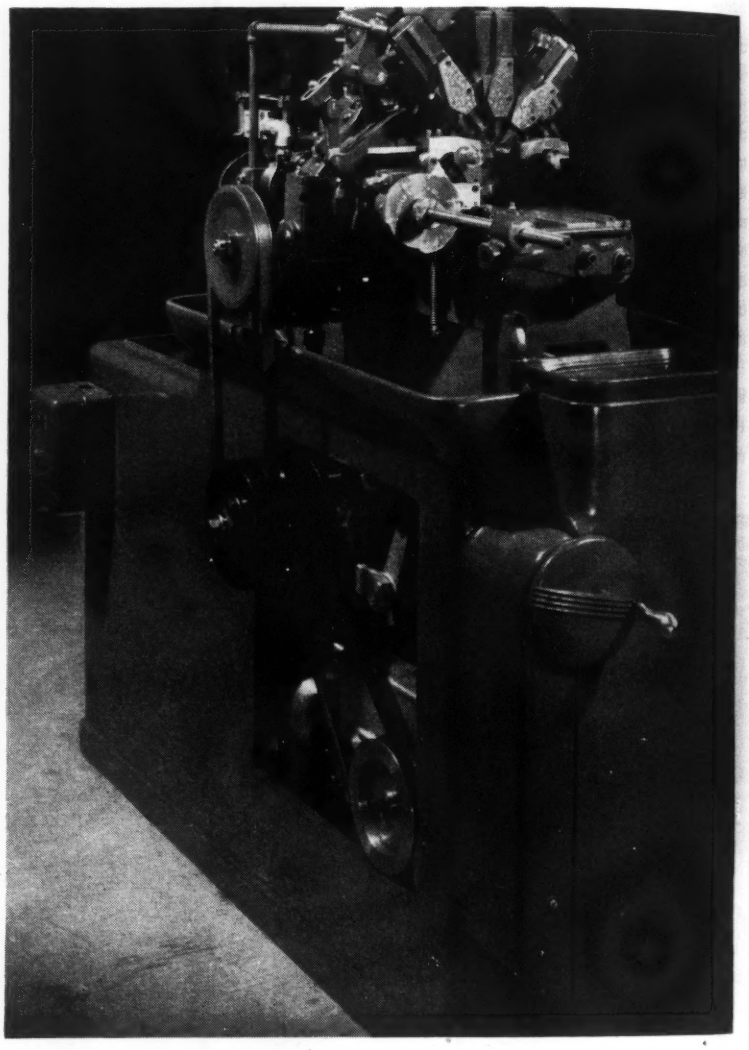


Fig. 6—Right—Another view of the machine shown in Fig. 3 illustrates the use of a pair of variable-pitch pulleys on a countershaft

ploying separate motors and transmissions, as in *Fig.* 3. In many other cases the motor and transmission are more conveniently applied when furnished as an integral unit, *Figs.* 2 and 4.

Generally the two pulleys in this type of drive are of equal proportions so that at high speed the transmission functions as a speed-up drive and at low speed as a reduction drive. For example, if the ratio of fast to slow output speeds is $n=6.25$, then the fast output speed is $\sqrt{n}=\sqrt{6.25}=2.5$ times the input speed and the slow output speed is $1/\sqrt{n}=1/\sqrt{6.25}=.4$ times the input speed. Since for constant input speed the horsepower capacity of the belt is approximately proportional to the belt speed, for constant input speed the maximum horsepower that can be transmitted at the high speed is 2.5 times that at the low speed. However, since torque output is proportional to the radius of the driven pulley, the safe torque at high speed is only .4 times as great as at low speed. These figures assume that the belt is loaded to capacity over the entire speed range.

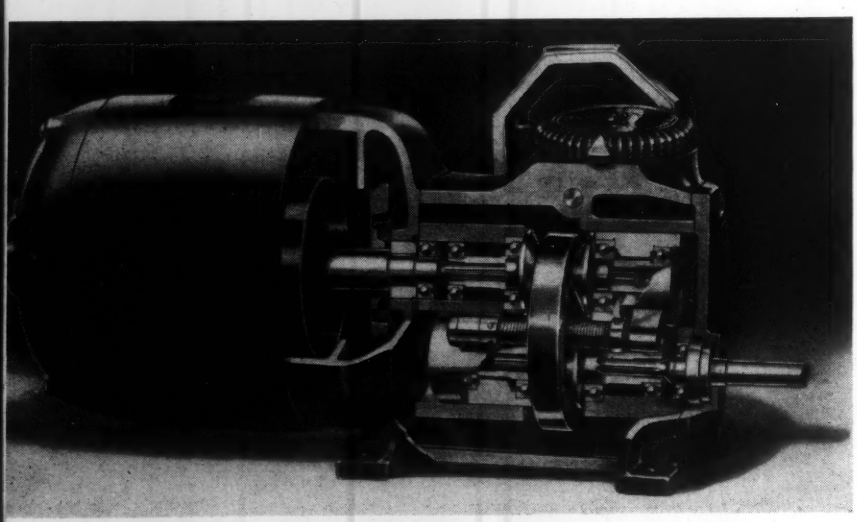
The foregoing characteristics may be modified somewhat by making the two pulleys unequal. Assuming, as an extreme example, that the high speed of the output shaft is made equal to the input speed, then for a ratio of 6.25 the slow output speed would be 1/6.25 or .16 times the input. If the pitch diameter of the driving pulley at slow speed is 1, that of the driven would be 6.25. For constant center distance the pitch diameter of both pulleys would be approximately $(1+6.25)/2=3.625$. Horsepower at high speed therefore would be 3.625 times that at slow speed while the torque would be $3.625/6.25=.58$ times as great. Such a drive, of which *Fig.* 2 is an example, therefore more nearly approaches constant torque capacity than the "over-and-under" drive previously considered.

To approach more nearly constant horsepower the fast output speed could be made equal to 6.25 times the input, then the high-speed horsepower would be $6.25/3.625=1.72$ times that at slow speed. Ordinarily, refinements such as indicated in the foregoing would not be justified and as a rule commercial standard pulley sizes are used, the unit being selected to cope with the most severe operating conditions likely to be encountered.

Large ratios without extreme changes in pitch diameters are made possible through use of a compound train, *Fig.* 5. Such an arrangement also permits lining up the driving and driven shafts, and facilitates adjustment of the belt tensions by providing for movement of the countershaft or jackshaft, as in the figure. Belt tension also can be automatically adjusted according to the speed ratio to compensate for the increased and decreased arc of contact.

A simple form of compound drive employs

Fig. 7—Below—Variable-pitch pulleys with metal ring provide compact all-metal drive



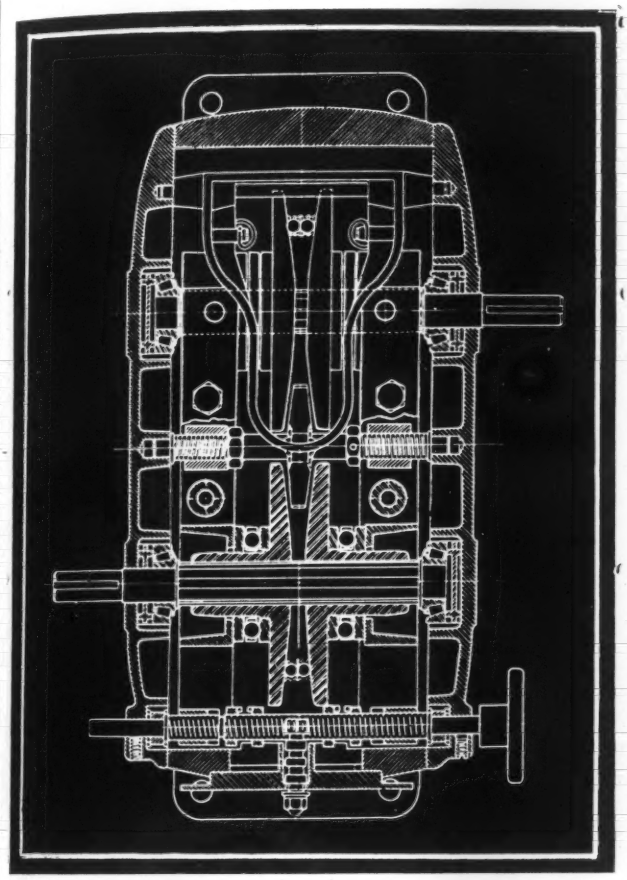


Fig. 8—Left—Side-contact chain has rollers to facilitate engagement and disengagement with variable pulleys

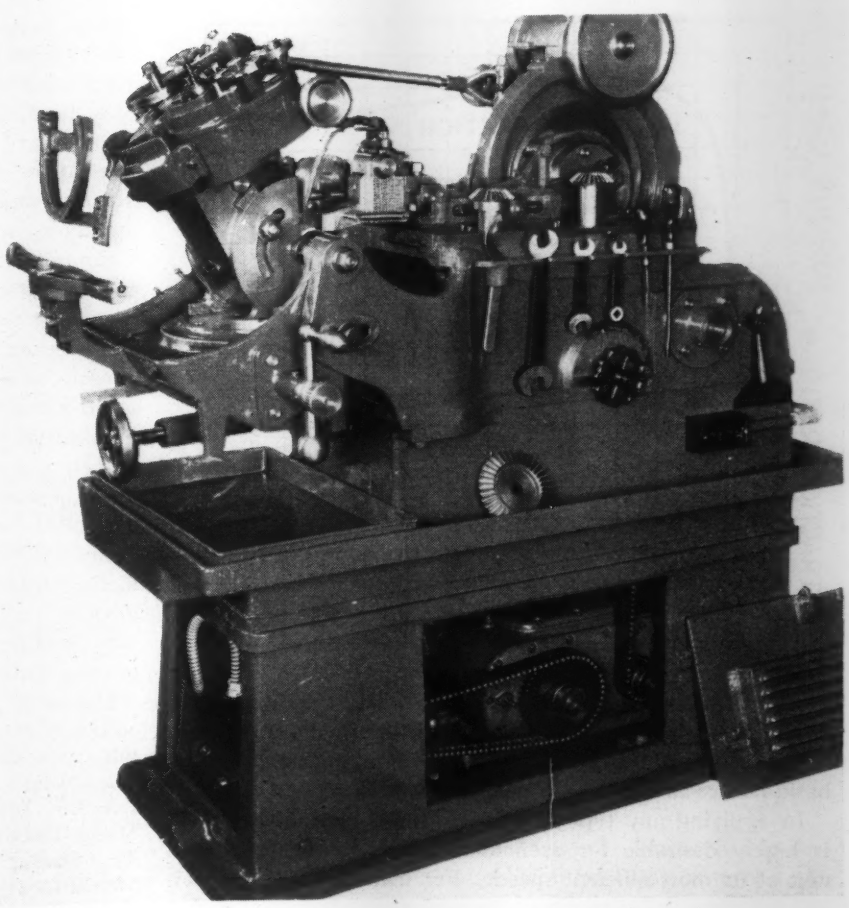
fixed-pitch pulleys on both driving and driven shafts, and an intermediate countershaft or jackshaft carrying variable-pitch pulleys. The countershaft is carried on a swinging arm so that the center distances between it and the driving and driven shafts may be changed. Inasmuch as the countershaft is located between the other two shafts, one center distance is shortened while the other is lengthened, the pulley cones adjusting themselves accordingly. *Fig.* 6, which is another view of the machine illustrated in *Fig.* 3, shows such a unit. This offers an interesting example of the use of two different types of transmissions for special purposes on the same machine. Employment of two countershaft pulley sets in series makes possible the attainment of quite high ratios.

In some double variable-pitch pulley designs a metal ring (*Fig.* 7) or, for light loads, a chain provided with contact rollers, *Fig.* 8, is used. Wedging action furnishes sufficient pressure to transmit the power under normal conditions. Where absolutely positive action is necessary, grooves are cut in the cone faces for engagement with a variable-pitch chain which, as it were, forms its own teeth from a series of hardened steel laminations or slats. Transmission units of this type are furnished either "bare" or fully enclosed, *Fig.* 9, for building in.

An entirely different principle is utilized in the transmission which is shown in the head illustration, Page 119, applied to the cam feed drive for an automatic screw machine. The unit is shown in cross section in *Fig.* 10 and consists essentially of a compound planetary gear train in which the nonrotating member is a traction ring engaging two tapered rollers at varying diameters, the ring being moved longitudinally to change the speed. The rollers are carried in a carrier ring which rotates at input speed, the necessary traction force between the rollers and the stationary ring being furnished by centrifugal force due to the mass of the rollers themselves plus that of a weighted block.

Performance of this type of transmission, as well as that shown in *Fig.* 7, depends on the basic traction curve depicted in *Fig.* 11. Torque which the unit is required to transmit at any time must lie within the region of usable traction indicated. However, should the torque exceed the limit, resulting in high slip, there is still considerable

Fig. 9—Below—Variable-chain drive unit, completely enclosed, is built into housing of gear-shaping machine

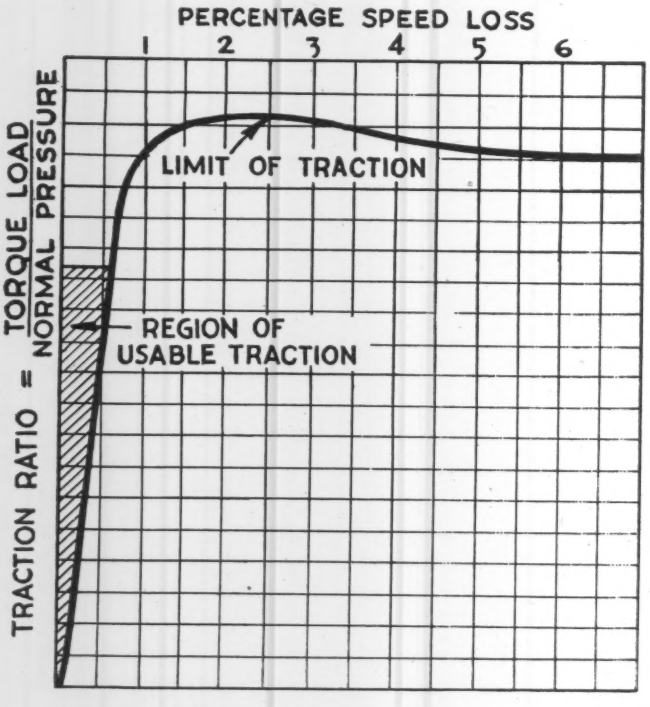


rolling because of the differential feature of the planetary system, *Fig.* 10, hence the danger of wearing the rollers in spots is obviated. With transmissions of this type practically any desired speed range can be obtained, including complete reversal. This characteristic, which permits "inching" or "jogging", also renders the provision of a clutch unnecessary.

In specifying any type of variable-speed transmission, as previously indicated, the primary considerations are the horsepower and torque requirements and the total speed variation. It is of equal importance to know the nature of the load and of the driving unit—whether steady, pulsating or shock—and whether frequent speed changes, stopping and starting under heavy loads, or load reversals are likely to be encountered in service. Such knowledge may influence the selection, as certain types of variable speed drive are inherently better adapted to absorb shocks than are others. As in all mechanical equipment, expected life and the type of service—normal (8 to 10 hours daily), continuous (24 hours) or intermittent—are basic factors in choosing a unit that will "stand the gaff".

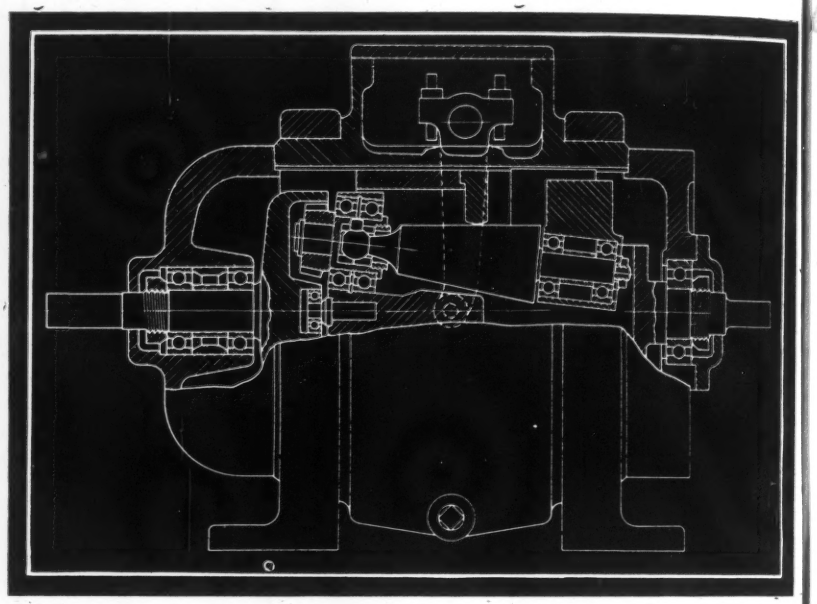
Fig. 10—Right—Planetary transmission with tapered rollers in place of one group of planet pinions provides speed changes down to zero and reverse

Fig. 11—Below—Basic traction curve governs performance of metallic rolling contact drives in the presence of oil



In applying any type of variable-speed transmission it is highly desirable for economic reasons to operate the unit at its most efficient speed. For many drives, therefore, some form of accessory drive such as gears, chains or belts is indicated.

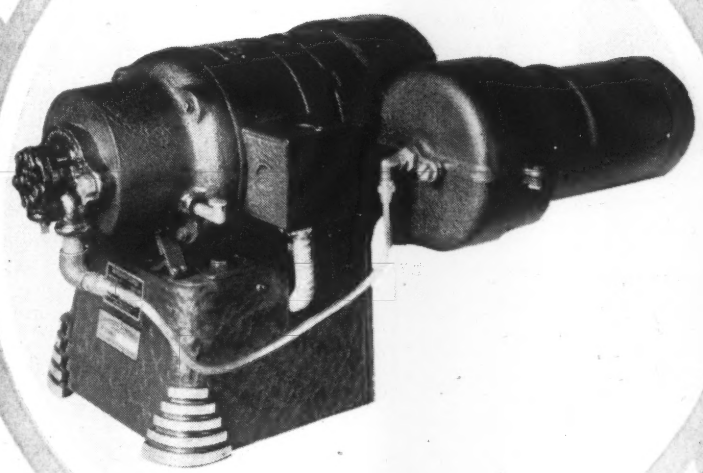
Inasmuch as input and output shafts on the unit may carry gears, sprockets or pulleys, the magnitude of the overhung load must lie within the capacity of the bearings provided in the unit. Likewise if worm, bevel or helical gears are included in the drive the transmission must be adequate to withstand the imposed thrust loads.



Although commercial units, particularly the enclosed types, usually are provided either with oil-impregnated porous bearings or packed antifriction bearings which will function without attention for considerable periods, it nevertheless is important to provide accessibility in the installation, permitting periodic inspection and lubrication. Unusual atmospheric conditions such as dust, moisture, fumes or extreme temperatures should be indicated when specifying, to insure that a unit incorporating the necessary protection will be furnished.

Controls for variable-speed transmissions are available in many forms—direct, remote and automatic. Manual controls, either at the unit or remotely, may be provided with vernier adjustment for accurate settings, and customarily are furnished with a dial indicating the output speed for a fixed input speed. Electric operation with pushbutton control and a speed indicator offers an extremely convenient method in certain applications. Inasmuch as practically any process or machine which can be controlled by a manually operated transmission may be better controlled automatically, the possibilities of automatic control by means of speed governors, tensioning devices, etc., should not be overlooked. Such controls may readily be adapted to most commercial units and often greatly increase their effectiveness.

MACHINE DESIGN acknowledges with appreciation the cooperation of the following companies in the preparation of this article: Allis-Chalmers Mfg. Co. (*Figs.* 3 and 6); Graham Transmissions Inc. (head illustration and *Figs.* 10 and 11); Lewellen Mfg. Co.; Link-Belt Co. (*Figs.* 8 and 9); The Master Electric Co. (*Fig.* 7); Reeves Pulley Co. (*Figs.* 1 and 4); Sterling Electric Motors Inc. (*Fig.* 2); U. S. Electrical Motors Inc.; Worthington Pump & Machinery Corp. (*Fig.* 5).



Which SMALL MOTOR To Use?

By C. T. Button
Holtzer-Cabot Electric Co.

ARMED with a knowledge of the inherent differences in behavior of the various motor types, and fully cognizant of the specific requirements of the mechanism to be driven, the machine designer should, with the aid of such quantitative data as percentage starting torque, etc., be able to exercise competent judgment in the proper selection of motors.

In dealing with the characteristics peculiar to different types of motors, it is necessary first to comprehend the principles which underlie the functioning of each. Al-

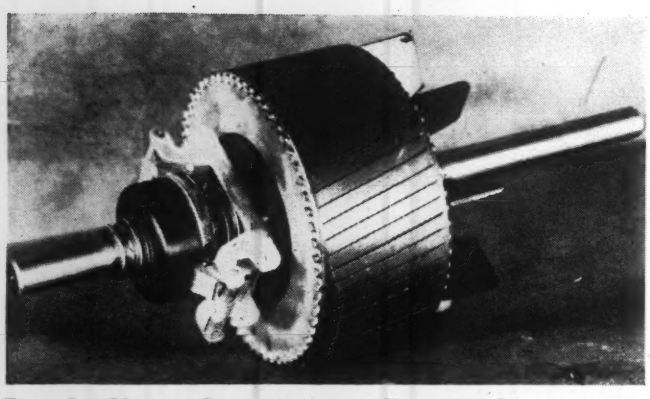
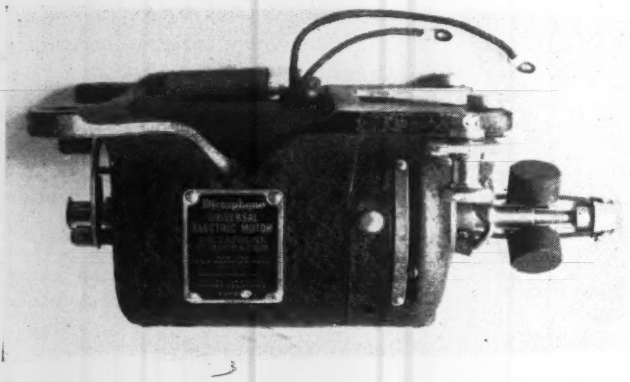


Fig. 1—Above—A typical small squirrel-cage rotor. Rotary part of centrifugal switch is at left side of rotor

Fig. 2—Below—Series motors offer high starting torque. Their speed may be controlled by mechanical governors



though such terms as "rotor" and "stator" are generally restricted to alternating current induction motor components, it is not improper to refer to the part which spins as the rotor and the part which is stationary as the stator, even though a direct-current motor be under discussion. For the present purposes, then, the term "rotor" will be broadly used, and qualifying adjectives will distinguish one type from another.

Undoubtedly the rotor type used in largest quantity today is the simple squirrel-cage unit. This is essentially a stack of annular laminations pressed on the shaft, with holes or notches around the periphery through which pass the conducting bars. The latter are joined together by end rings so as to form a "squirrel cage" as illustrated in *Fig.* 1. The conducting material for the bars may be copper, bronze, brass, or aluminum alloy.

When placed in a rotating magnetic field which traverses the rotor in a plane perpendicular to the shaft, rotor bar currents will be induced which produce torque. If this torque exceeds the resisting load, the rotor will accelerate and attempt to catch up with the angular velocity of the rotating field. It never does so unless irregularities are introduced in the rotor construction. If these irregularities have the effect of producing salient rotor poles (or polar distribution of flux density) the rotor may pull into step with the rotating field, the result being a synchronous motor.

In a simple squirrel cage motor, the torque created in a given size rotor will be dependent upon: (1) Flux magnitude; (2) "slip" or difference in angular velocity of rotor and field; and (3) rotor resistance.

Permanent differences in rotor resistance are obtain by changing cross section or material of the conducti bars or end rings. Controllable differences in rotor r sistance may be obtained by using a wire-wound rot connected to slip rings, and by introducing resistance the rotor circuit externally. This wound-rotor constru tion is widely used in large polyphase motors. While is not a common type in small alternating-current moto it merits consideration for the sake of theoretical co pleteness.

Induction motors may be classified on the basis of rot construction as follows:

1. **Squirrel Cage**
   - (a) Nonsynchronous
     - Low resistance rotor
     - High resistance rotor
   - (b) Synchronous
     - Flats, notches, slots or other polar irregularities (r luctance type)
     - Permanent magnet partial section (hysteresis typ

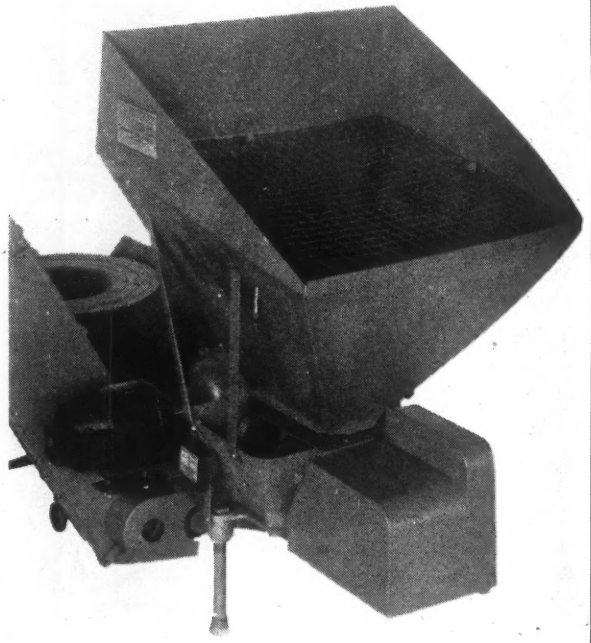


Fig. 3—Stoker driven by repulsion-start induction motor which has starting characteristics similar to series type

2. **Wound Rotor**
   - (a) Nonsynchronous, variable external resistance
   - (b) Synchronous, with D.C. introduced through slip rings
3. **Special types,** such as non-self-starting clock motors of reluctance type, permanent magnet (or hysteresis) rotors, and inductor types. Frequently some of these are so-called subsynchronous motors.

In regard to the characteristics of squirrel cage motors, a general statement may be made that high-resistance rotors have high starting torque and high slip (and usually higher running losses) whereas these quantities are low for a low resistance rotor. High starting torque is an advantage, but high slip is usually a disadvantage.

Various types of squirrel-cage induction motors may be listed as follows:

1. Shaded pole
2. Permanent split capacitor
   (also referred to as low torque or single value capacitor)
3. Split phase
4. Capacitor start
5. Capacitor start and run
6. Two or three phase.

If polyphase alternating current were universally available there probably would be little reason for existence of the first five motors listed, which are all single phase. A single-phase motor with a strictly single-phase winding may be compared (roughly) with a single-cylinder reciprocating steam engine, which must be pushed off dead center and accelerated to a certain point before it keeps on going. A positive torque impulse is received twice per cycle. (This is perhaps the principal noise-producing factor in single-phase motors).

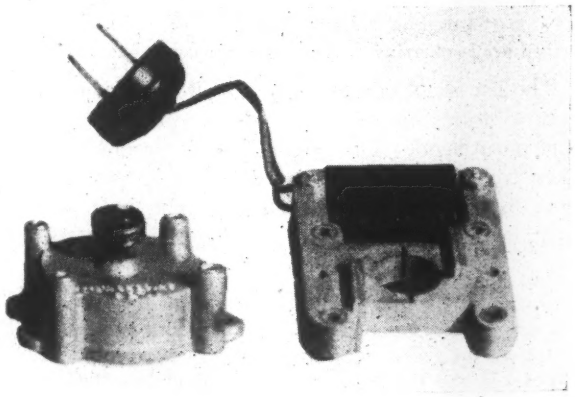
No One Type Is Ideal

Naturally if any one type of single-phase motor had all the advantages it would displace all the others. The first five types of squirrel-cage induction motors listed in the foregoing may briefly be classified as follows:

	Advantages	Disadvantages
1.	Low cost Simplicity	Low torque Low efficiency
2.	High efficiency Simplicity Quietness Susceptible to speed control	Low starting torque
3.	Low cost (in ratings where performance of shaded pole motor is too poor to be used)	Requires centrifugal switch Starting torque limited
4.	High starting torque	Requires centrifugal switch
5.	High starting torque Quietness.	High cost.

All of these five single-phase motor types may be made synchronous if provided with suitable rotors. Types 1 and 2 may be used where variable speed is desired, as for fans in particular. This is accomplished by reducing the strength of the motor by various means, such as series resistance, autotransformers, or changes in connections to stator winding. Types 3, 4 and 5 generally cannot be used with speed control because of the centrifugal switch required to disconnect the starting winding and/or starting capacitor.

Fig. 4—Shaded-pole motors are used to drive synchronous clocks. Also to energize small fans, valve controls, etc.



Reverting to a consideration of basic rotor types, the wound armature motor with brushes and commutator has not yet been mentioned. This motor is not an induction type since the rotor is conductively excited (except for the repulsion type which is discussed below).

The series type motor with wound armature and commutator does not require a rotating field. Its principle of operation is the same as that of a series direct-current motor. Not all series direct-current motors will run properly on alternating-current, however, as certain design modifications must be effected to minimize alternating-current reactance and also to secure passable commutation. The merits and shortcomings of the series motor may be listed as follows:

Advantages	Disadvantages
May be made universal for operation on A.C. or D.C.	Not constant speed
Susceptible to speed control	Commutation on A.C. limits life—usually not suitable for strictly continuous duty
High starting torque	
Fair efficiency.	Causes radio interference
	Cost.

Also warranting consideration is another wound-armature type, the repulsion-start induction motor, a typical

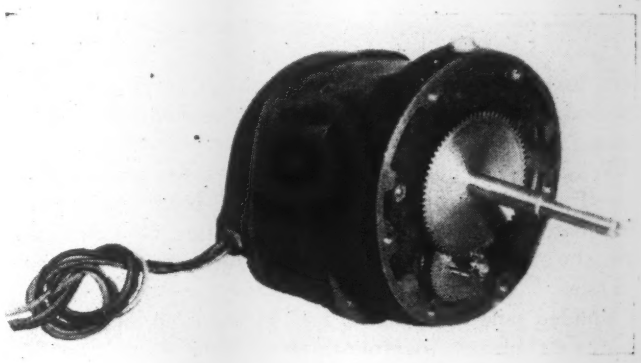


Fig. 5—Capacitor-type synchronous gear motor which replaces the shaded-pole type when high output is required

application of which is illustrated in *Fig.* 3. When running, it is an induction motor, its operation being comparable to a squirrel-cage motor. There is no essential distinction except that the end ring for the squirrel cage is replaced by short-circuited commutator segments. On starting, the repulsion characteristics are similar to those of a series motor. Instead of taking line current through the brushes, however, the latter are short-circuited and the armature may be regarded as the secondary of a transformer in which the stator is the primary. Thus the brushes are shorted on starting, and the whole commutator is short-circuited when the motor comes up to speed. The repulsion-start induction motor has in large measure been replaced by the capacitor-start motor which per-

haps is equally dependable for all but extremely difficult starting jobs and probably is lower in intrinsic cost. Any of the motors mentioned may be equipped, of course, with a speed reducer to provide power economically at low speeds.

Sometimes relatively small factors determine what motor should be used. For example, a domestic sewing machine has intermittent duty, requires varying speed, must be moderate in cost, have high starting torque, and offer considerable power in a small space. The series motor serves well here and incidentally makes it unnecessary to worry about whether the home is in a direct-current or alternating-current district.

Vacuum cleaners also use series motors for similar reasons except that high speed rather than varying speed is desired. High starting torque is not needed but does no harm.

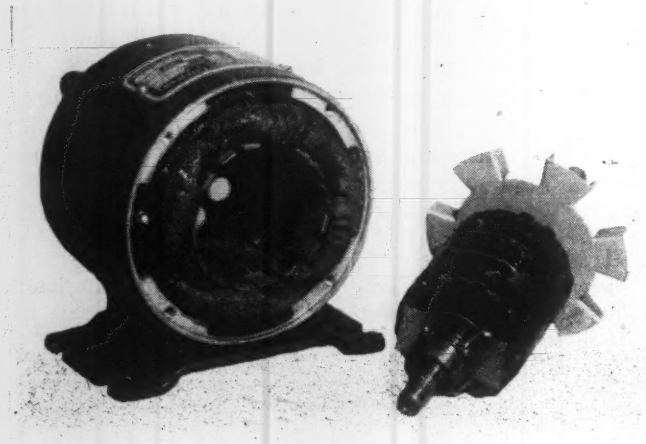


Fig. 6—Stator and rotor assemblies of small split-phase synchronous motor. Note rotor fan and centrifugal switch

Washing machines and refrigerators require constant speed and high power output. Split-phase motors are adequate for the former, but the addition of a capacitor for starting purposes is required for refrigerator compressors.

Shaded pole motors (see *Fig.* 4) are adequate for such things as furnace damper motors, some valve controls, small fans and similar miscellaneous jobs. They also are used for synchronous clock motors. Where more output or less heating is desired, as in recording instrument chart drives and large fans, the shaded pole motor gives way to the capacitor type, either synchronous (see *Fig.* 5) or nonsynchronous as the case may be.

Motor parts such as bearings and housing are frequently built in as integral parts of the driven device. Such was the procedure in the case of the oil burner shown in the illustration which heads this article. In this case the motor was adapted for direct mounting of the fuel pump. Again, fields and armatures of universal motors are so installed in cash registers in a manner comparable with that in which shell type polyphase motors are applied in some machine tools.

Drink mixers, portable tools and office appliances also use universal motors. Adding machines requiring the motor to start from standstill each time the motor key is pressed generally use series wound motors, usually with a governor. On the other hand large tabulating machines use heavy-duty continuous-running induction motors.

Various types of motor mountings (resilient base, flange, etc); degree of enclosing for the motor frame; selection of the best bearings for the job; possibilities of hazardous gases, dirt and humidity, are all details which should be considered before the final motor requirement specification is completed.

Future Developments

Potentialities for future developments, particularly in the control field, are all but limitless. The following is but a brief outline of what such developments might be:

1. Adjustable speed alternating-current motor with shunt (or essentially constant but adjustable speed characteristics. This result may be accomplished with or without rectifying means and/or electronic tubes.

2. Spinner or drag cup motors with low inertia. These motors have neither squirrel cage nor wound armature, the rotor consisting merely of a light thin cup which may be made of aluminum. The essential virtue of these motors is extremely low rotor inertia for quick response. A secondary virtue is smooth torque as there are no rotor slots or bars.

3. High frequency motors. Wide use of 400 cycles on aircraft may possibly be carried into some special commercial fields where high speed, light weight and small size for the horsepower delivered are desired. The use of 180 cycles industrially is not new.

4. "Selsyn" or position-indicating motors.

5. Torque motors are not new, but perhaps have not been as widely used as might be. This motor may be designed to exert continuous torque while not running, or the rotor may move through only a small angle. It has been compared to a spring of infinite length. The torque exerted may be regulated by voltage control.

6. Control auxiliaries and accessories such as:

(a) Electronic controls including phototubes
(b) Amplidynes or other modification of Ward-Leonard type
(c) Thermal overloads and other thermal limits and controls
(d) Carbon piles
    Voltage regulators Speed regulators
    Remote control of resistance for various purposes
(e) Governors
    Contact making—constant speed
    Contact making—adjustable speed
    Saturable reactors
(f) Rapid-reversing motors and servo-mechanisms.

The list is not complete, being intended as only suggestive. Various combinations of controls and motor types may be utilized. The possibilities of producing power or force from electrical energy governed in any pattern desired in response to conditions of light, heat, mechanical position or condition, pressure, electrical or other quantities are infinite.

The author is pleased to acknowledge the collaboration of the following companies: Dictaphone Corp., *Fig.* 2; Iler Laboratories, Inc., *Fig.* 4; Bodine Electric Co., *Fig.* 6; Holtzer-Cabot Electric Co., photograph heading the article and *Figs.* 1, 3 and 5.

Built-in Automatic Temperature Controls

By V. G. Vaughan
Spencer Thermostat Co.

LATE IN 1943 there was a railroad wreck at Philadelphia, with considerable loss of life, which was attributed to an overheated bearing. It is to be expected that the possibility of this type of wreck will someday be eliminated by the use of a bearing with a properly applied built-in temperature control.

To attempt to describe all of the potential applications for built-in temperature controls would require at least one good-sized text book, if not several volumes. In this short article, the discussion will be confined to a few applications now in regular

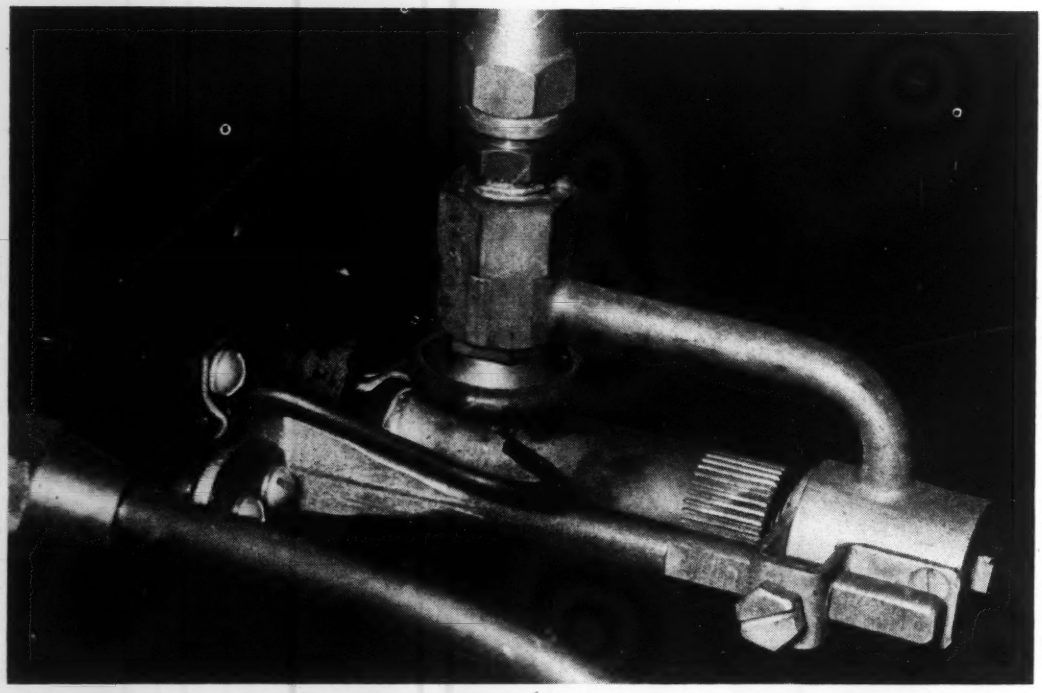


Fig. 1—Left—Safety control for flow of gas to refrigerator burner utilizes a bimetallic thermostat which operates the main gas valve

Fig. 2—Below—U-shape thermostat for gas oven burner closes main gas valve in case of pilot flame failure

production and will hazard a forecast as to some of the near-future applications.

Built-in automatic temperature controls may be mechanical thermostats, of which the automobile water-circulating thermostat shown in the head illustration of this article is the one most intimately connected with our everyday life. It has been used in all automobiles and trucks built in the past ten years. Mechanical types may control the flow of liquid or gasses. Such an application is shown in *Fig.* 1 for a gas refrigerator. In event of pilot flame failure, a bimetal thermostat-controlled valve shuts off the main gas supply. A U-type thermostate operates similarly for a gas oven as shown in *Fig.* 2. Another mechanical control is illustrated in *Fig.* 3 for motor heating units. If the temperature exceeds a safe value the valve opens, obviating any possible damage from excessive pressures. There have been some attempts also to control the temperature of friction clutches by mechanical means.

Most Controls Utilize Electric Circuits

The greatest variety of built-in temperature controls are those incorporating electrical switches and functioning through the medium of an electric circuit. A common example is the thermostat in the automatic electric iron.

All known types of temperature-responsive mediums have been used in the design of built-in temperature controls. Some basic forms are:

1. Expanding rods or wires
2. Expanding liquids
3. Expansion due to vaporization of liquids
4. Bimetals
5. Resistance change
6. Thermal emf (thermocouple).

There are many individual designs depending on the type of thermal-responsive element used, on the machine into which it is to be built, and on the individual designer's concept of the proper solution of his control problem. Typical shapes of bimetal elements are shown in *Fig.* 4 to indicate the variety of forms of this type alone. Representative disk-type thermostats, having electrical contacts, are illustrated in *Fig.* 5.

Built-in automatic temperature controls may operate for a full cut off and cut on of power, or they may be of the throttling type. Control may respond to a temperature change at the point in the machine where it is desired to control the temperature or it may anticipate tempera-

ture changes in the machine by being responsive to both the temperature in the machine and the source of heat, causing temperature changes in the machine. It usually is impossible to control the temperature of all parts in a machine and, therefore, some point must be selected as the spot at which the temperature is to be controlled.

Automatic control of temperature will always cut out and cut in. In doing this, it may cut off all or only part of the load. A temperature-limiting device, however, may automatically cut out and cut in, or it may automatically cut out when a dangerous temperature is reached and require a manual operation to restart.

While there are many built-in temperature controls in use, it is surprising how many machines are designed without provisions being made for the installation of a temperature control. The result is that the performance of many machines is limited by the field application of some external device.

Fig. 3—Above—Built-in type of temperature relief valve. If temperature exceeds a safe value the valve opens. It closes automatically when operating temperatures are reached

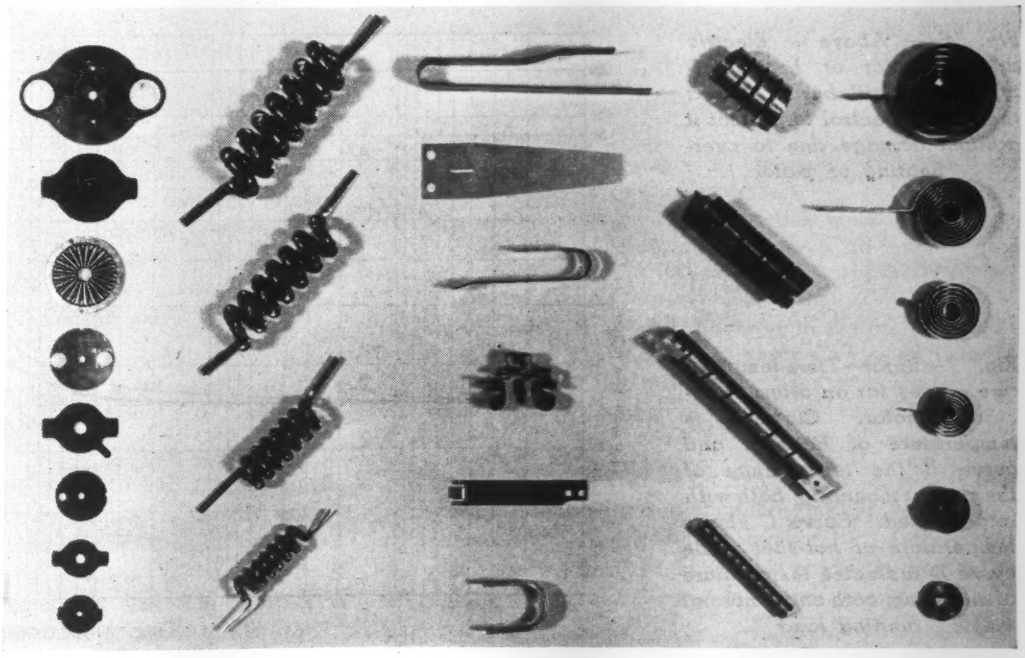


Fig. 4—Right—Various shapes, sizes and kinds of bimetal thermal elements for built-in temperature controls

Electric motors are a good example with respect to comparing the performance characteristics of external temperature controls versus built-in temperature controls. The history of the electric motor and its control is filled with many changes denoting progress and improvement in design and application of controls for temperature protection. As most motors overheat due to an excessive flow of current, it has been common practice to protect the motor from excessive temperatures by the use of an external thermal device (fuse, thermal cut-out, thermal relay, etc.) operating from the current in the circuit.

These devices when properly applied will protect the motor, but in many applications the protective device so limits the output of the motor under normal conditions that it invites the installation of higher ratings which allow the motor to run at higher temperatures when overloaded, with a resulting burn-out of the motor under abnormal conditions.

Protecting for Ambient Temperatures

An application in which this system may happen is a motor running a rotary type oil burner, *Fig.* 6, with the control mounted in a box on the wall near a window. In this case the heat of the burner has the effect of reducing the capacity of the motor, necessitating the use of a higher rating control. In cold weather the cold ambient surrounding the control will have the effect of increasing the rating of the control, with the result that if an overload occurred under these conditions, the motor probably would not be protected from overheating.

The way to overcome this is by having the combination of motor and temperature-control device arranged so that they are both responsive to the same ambient temperature changes. This can only be accomplished by

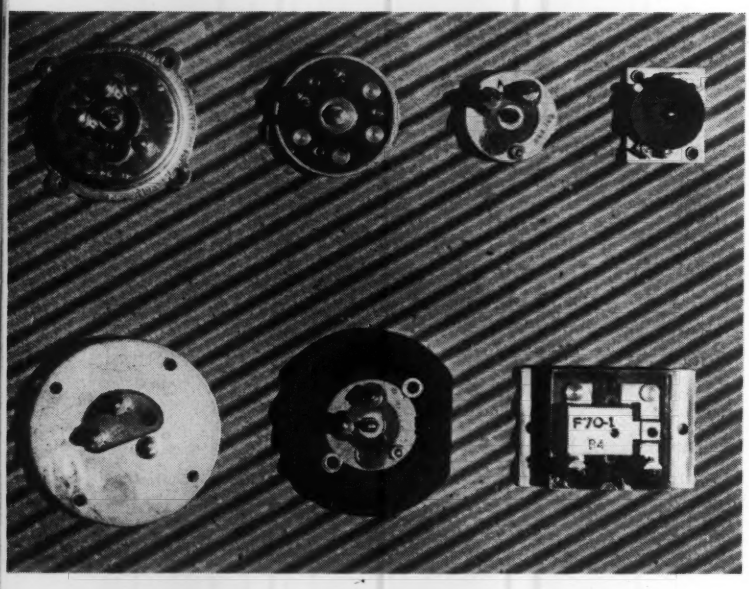


Fig. 5—Above—Bimetal-disk temperature controls utilizing electrical contacts

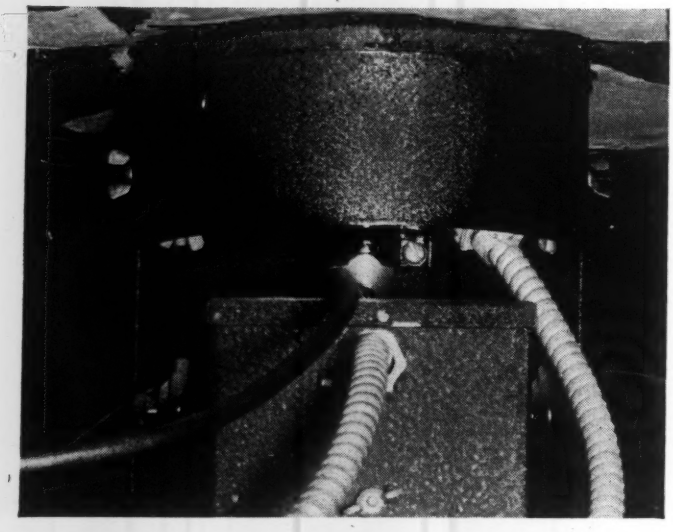


Fig. 6 — Above — Electric driving motor of horizontal-type rotary oil burner has a thermostat control to protect it against damage due to overheating of motor

Fig. 7—Right—Time-temperature curves for an alternating-current motor. Curve A is temperature of hot-spot and curve B the temperature of thermostat mounting, both with rotor locked. Curve C shows temperature of hot-spot while curve D indicates temperature of mounting, both under normal running load

designing the motor with a built-in control.

Most designers appreciate how such a control can be applied to a device where the control responds to the temperature of the medium being controlled, as in the automobile water-circulating thermostat. Here the control is immersed in the liquid whose flow it is to control. There is not the same universal appreciation of the solution of a problem where the control must be responsive to a combination of the origin of the heat and its result, such as is accomplished by the built-in temperature control in the electric motor.

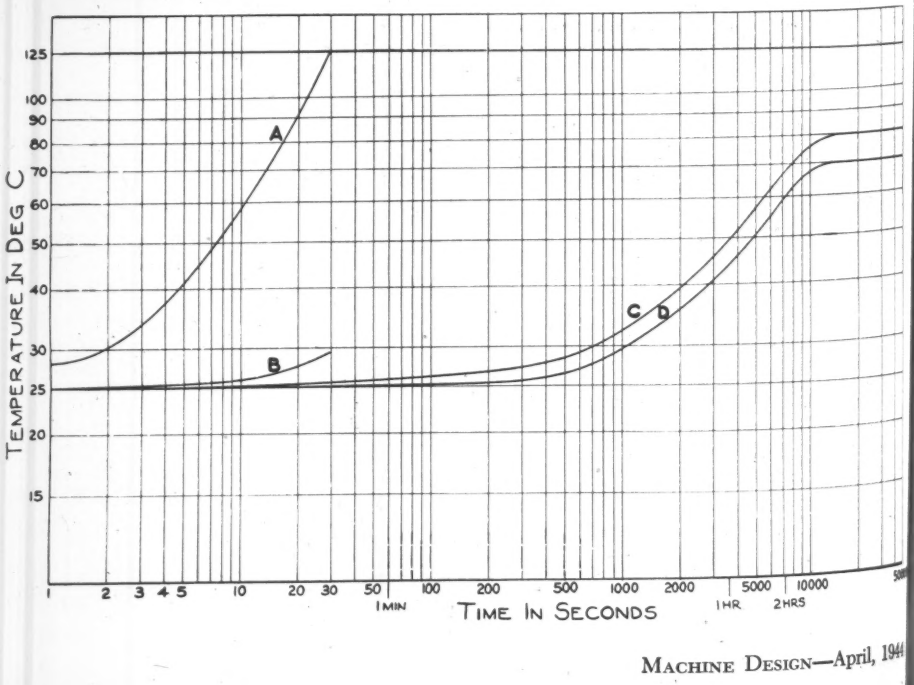
Control Is Tailored To Fit

In working out the proper design of a given motor, the windings must be designed to fit the physical limitations of the motor frame and at the same time furnish the torque and speed characteristics desired. In other words, the winding must be *tailored* to fit the application.

It would, therefore, seem logical to tailor the temperature control to fit the heating characteristics of the motor. When connected to the circuit, an electric motor does not have a uniform temperature. Instead it has hot-spots and usually these are the weak thermal parts of the machine and, therefore, are the places that must be protected. The hot-spot is usually in the center of the winding located in the slot and the temperature control is located as near as possible to this spot.

The temperature at the mounting of the control naturally lags behind the temperature of the hot-spot as indicated in *Fig.* 7, curves *B* and *D*, where the relative rates of heating are shown for the hot-spot and the temperature of the mounting of the built-in control in an alternating current motor running normal and with the rotor locked.

It is to be noted that when the rotor is locked, the temperature at the control, curve *D* has barely changed from room temperature when the hot-spot, curve *A*, has reached the dangerous temperature. This means that



the source of heat (the current) must be used to boost the temperature at the control so that it prevents the motor from overheating when locked and at the same time allows the motor to carry maximum safe load when running. This is accomplished by building a heater into the temperature control. The curves in *Fig.* 8 of an aircraft motor illustrate the versatility of control for motors which are subject to a wide range of ambients, loads, ventilation, voltage and altitude.

May Reduce Load on Drive

In most of the present-day applications to electric motors it has been common practice to shut down the motor or to give an alarm so that the operator can reduce the load. As the field of application broadens there will be occasions to have the temperature control operate mechanisms which automatically will reduce the load or increase the cooling rate of the machine so that the motor can continue to carry the load without being damaged and without stopping the machine.

There are a number of advantages to be gained in using built-in temperature controls:

1. First and foremost is the fact that the machine designer has complete control of the design of the machine
2. When the machine with its temperature control is tested at the factory, it insures proper testing of the combination and results in a more uniform product, giving precision uniformity which is seldom attainable with a field-applied temperature control
3. The more uniform the product a manufacturer can make, the less the excess margin over normal requirements the machine designer must allow in his machine, and this will always result in a decrease in the overall unit cost to the customer
4. The machine designer can be assured of dependable operation, for he gets his life test on the built-in temperature control at the same time that he makes his life test on the machine.

Each machine designer is likely to look for progress only in his own special field. Many take care of improvements only as the customers demand the incorporation of these improvements. History is filled with incidents that illustrate such inertia. Having used the automobile before as an example, we might turn to it again. The original automobile was designed and sold without tires, lights, top, bumpers, thermostats, etc. It was not the automobile designer that made the first applications of these necessary built-in features. When the items were first introduced, the customer had to buy them as accessories and have them fitted to his car. The result was:

1. Relatively poor performance
2. Lack of uniformity
3. Undependable operation
4. High unit cost.

The customer in many cases seems to find these things out before the machine designer, and it is usually customer pressure through the medium of competition that brings universal acceptance of built-in features. The designer can anticipate his customer's demands by keeping alert to the changing events in fields allied to his own.

Because of the engineering and tool costs necessary to convert existing designs to incorporate temperature controls, applications have been somewhat limited to large-volume items. This need not hold true in the future, for the modern designer appreciates that it costs no more to design and tool initially for the machine with provisions for a temperature control. This will be accelerated as more design knowledge and more standard designs of

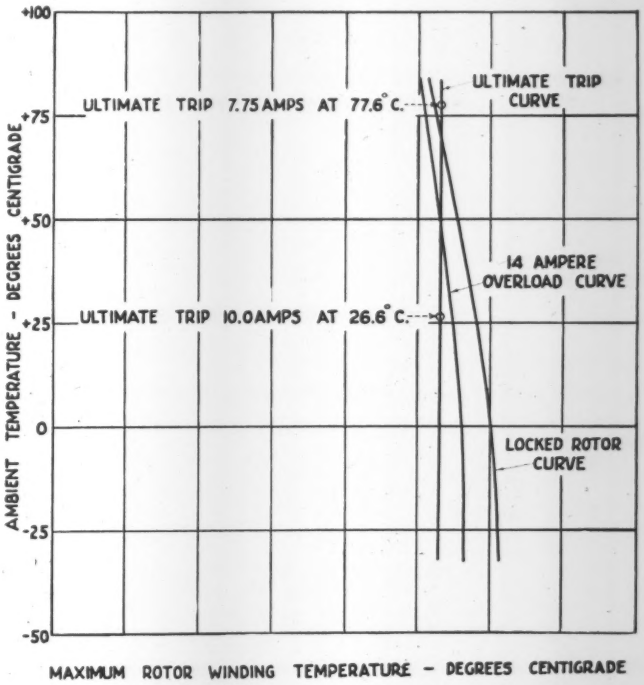


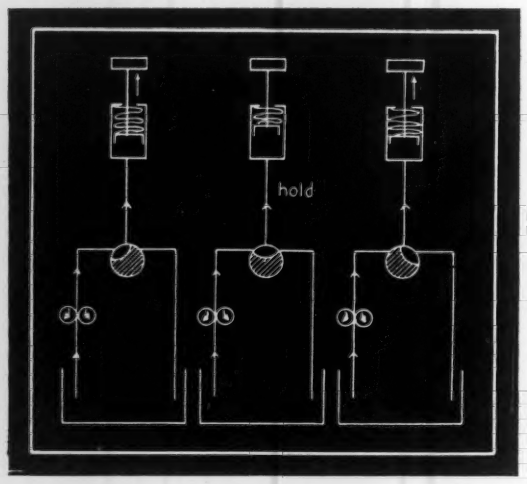
Fig. 8—Performance data on 1/8-horsepower, 24-volt, continuous-duty aircraft motor with inherent overheat protector. Full-load current for this motor is 6.8 amperes

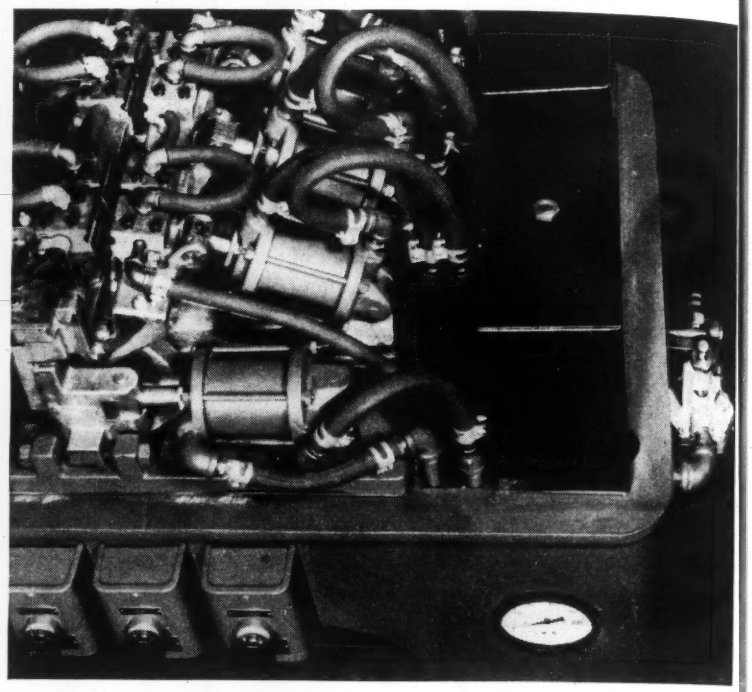
controls become available.

In the near future we can expect a greater increased use of built-in temperature controls in the following designs:

1. Electric motors
2. Transformers, particularly in specialty fields such as electric ignition for oil and gas
3. Both gas-burning and electric heating units will have an expanded use of temperature controls to improve the automatic lighting of gas and the temperature control of units
4. Electric refrigerators will shift over to the use of a built-in control on the dome of hermetically sealed units to insure better overall performance with greater safety
5. Air-conditioning equipment will have all limit controls built in.

This list could be carried on indefinitely. It is best that we leave it here with the thought that we as machine designers must be more alert to temperature problems of the machines we design and at least study the possibility of built-in temperature controls before completing and tooling our designs.





Control Criterion for Hydraulics

By R. S. Elberty
Consulting Engineer
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MACHINE designers have recognized that control functions may be obtained in a number of ways and that hydraulic and pneumatic controls have been developed to a point where their complexity compares with advanced electrical controls. A check on the design of these complex hydraulic and pneumatic controls will be of help to the designer who considers control a general problem to be solved in the best manner, whether it be electric, hydraulic, pneumatic, or mechanical.

The control criterion, originally developed for electrical circuits, states that there can be as many control functions as there are electrical contacts in that circuit if they are restrained contacts; and there will be one more function if the contacts are all free. The development of the criterion* considered any combination of series and parallel contacts, and there is nothing in hydraulic and pneumatic control circuits to require any additional considerations. Circuits using fluid pressure are controlled by valve ports which are comparable to the contacts used in electrical circuits. Substituting ports for contacts, and considering all the conditions of fluid pressure control, the control criterion then may be applied equally well.

The action of hydraulic fluids is much the same as that of electricity. When a port is open to pressure, power flows to the operated member. Closing the port, however, differs from opening an electrical contact, since electricity leaves the conductor, while the hydraulic fluid still fills the tubing. This is the major difference that must govern the analysis of the functions of control by fluid pressure. Also, it takes two ports to establish a circuit.

For example, a hydraulic press might use a spring return. It may be controlled by a simple valve with three ports. One port connects to the cylinder, one to pressure, the third to exhaust. With the cylinder port open to pressure, the pressure stroke is actuated. Opening this port enables the hydraulic fluid to hold the press in position against the action of the return spring, and closing the port to exhaust permits the spring to return the press to the open position. The port connected to the operated member was used twice and in many types of valves this would actually be two ports. Whatever the construction, this port must be considered as a quantity of ports. The valve provides three functions, and the relation between this and the electrical circuit is clear. But the ports must be considered in groups of two and the criterion becomes

$$F=\frac{P}{2}-1$$

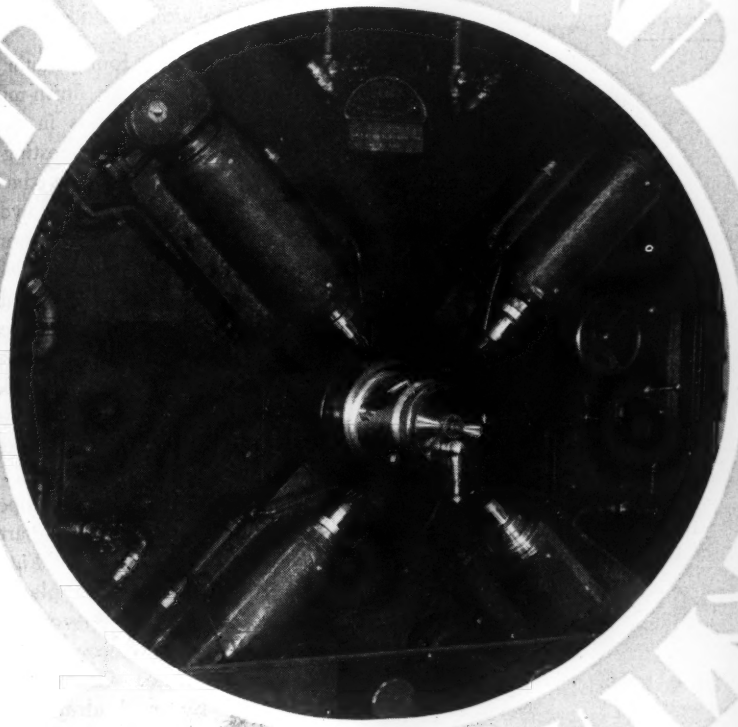
The number of functions equals one-half the number of ports plus one.

Suppose the valves were made to have a spring action so that the first port was open either to pressure or exhaust, but was never closed. This adds a restraint to the valve and eliminates the condition where the press is locked in the closed position. The two general expressions for the criterion may then be written:

$$F=\frac{P_f}{2}-1 \quad ; \quad F=\frac{P_f}{2}-\frac{P_r}{2}$$

Where F = number of functions, and P_f = number of free ports, and P_r = number of restrained ports.

*Development of this criterion is discussed in "Control Criterion—A Check on Design" and "Restraint of Contacts", MACHINE DESIGN, April and May, 1943, respectively.



Combined Methods Give Best Results

By Wm. C. Oberem
Chief Engineer
Snyder Tool & Engineering Co.

WITH mechanical, hydraulic and electrical drives and controls to choose from, the designer of special-purpose war production machines is in the fortunate position of being able to satisfy the most exacting specifications covering automatic and foolproof performance. Through a combination of these different types of drive and control methods, operation of the machine illustrated in *Fig.* 1 is reduced to pushbutton control and a simple loading and unloading procedure.

Design requirements may briefly be set forth by outlining the job performed:

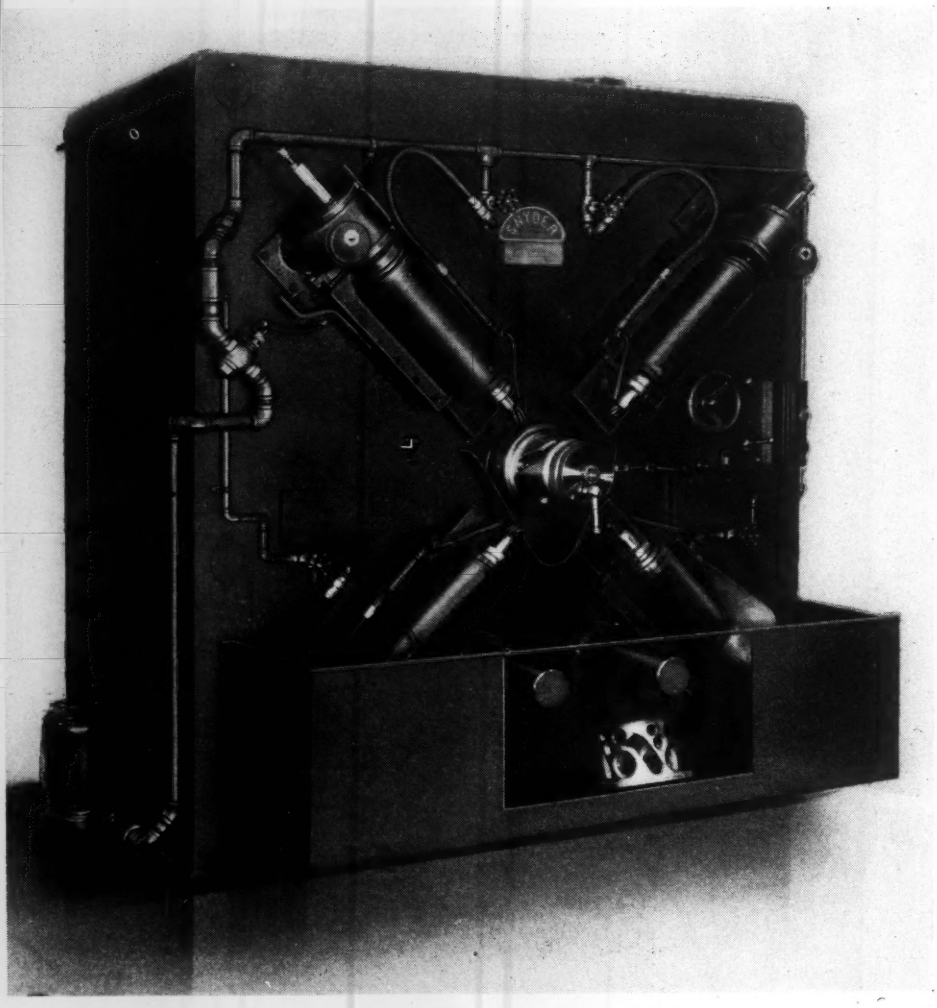
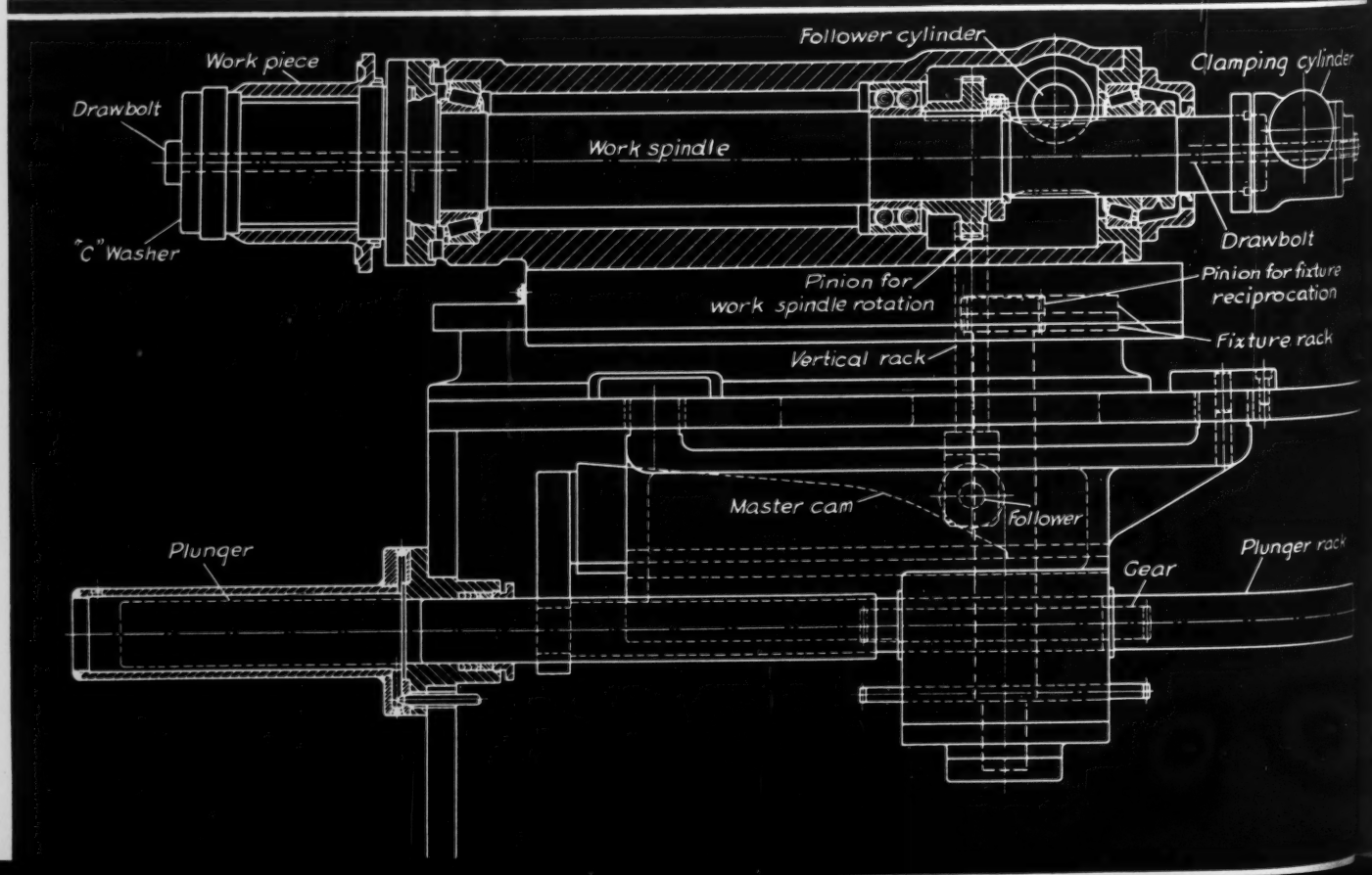


Fig. 1 — Left — Special-purpose milling machine finish-mills four slots in aircraft propeller cam in seven minutes

Equipped with four motor-driven spindles, the machine finish-mills simultaneously the four slots in the aircraft propeller cam shown resting on the splash pan at the front of the machine, *Fig.* 1. The workpiece is hydraulically rotated and reciprocated to form the cam contour while the mechanically interconnected spindles are hydraulically fed into the work. All adjustment points are enclosed within the machine to prevent tampering. Control of the machining cycle is electrical, the entire operation requiring approximately seven minutes.

Consider first how the proper reciprocation and rotation are imparted to the workpiece. As shown in *Figs.* 1 and 2 the workpiece is held in place in the fixture by a "C" washer, secured by a long drawbolt passing through the center of the work spindle. Clamping pressure is furnished by a hydraulically-actuated plunger at the back of the machine, *Fig.* 2, which tightens the drawbolt through a cam. This is shown diagram-

Fig. 2—Below—Work fixture is reciprocated by hydraulic plungers and rotated by cam, rack and pinion mechanism to give proper slot profile



matically at the left center of the hydraulic circuit diagram, *Fig.* 3. Clamp handle indicated on the diagram operates a valve which controls flow of oil to the clamping cylinders. Incidentally, a switch connected with this handle opens the electrical circuit controlling the cycle except when the workpiece is clamped in place, obviating any danger of prematurely starting the cycle.

Reciprocating the Workpiece

Longitudinal motion or reciprocation of the work fixture is imparted by means of two hydraulic plungers indicated at lower left of *Fig.* 2 and at the right of *Fig.* 3. These plungers are connected to racks which mesh with a common gear. A pinion rotating with this common gear engages a rack attached to the fixture slide. Diameter ratio between these two gears is four to one, so that the fixture slide moves only one-fourth inch for each inch of plunger movement. The reasons for using this mechanism instead of driving the fixture table direct from the cylinder are two:

a. Mechanical advantage enables use of smaller cylinders and lower pressures.

b. Accurate control is easier since cams and dogs for tripping valves and limit switches move four times as far as the work itself.

Thus, dimensional accuracy of .004-inch on these control elements results in .001-inch accuracy of work. Each rack is split longitudinally at the middle of the tooth space, adjustment being provided for slightly staggering the teeth in each half, thus entirely eliminating backlash.

Rotating the Workpiece

Rotation of the work fixture is effected through a reciprocating master cam attached to one of the plungers. Shown in *Fig.* 2 and indicated schematically at the right of *Fig.* 3, this cam engages a reciprocating roll follower which, in turn, rotates the work fixture spindle through a vertical rack and pinion. Inasmuch as the cam moves in one direction at four times the speed with which the fixture slide moves in the opposite direction, the relative speed is five times that of the work and the cam therefore must be five times as long as the stroke of the work fixture. As a result, the action of the machine and cutters in the work is exceptionally smooth.

Inasmuch as cutting occurs on both strokes of the work fixture, it is necessary that the follower which provides rotation be maintained in contact with the master cam. This restoring force is furnished by a hydraulic cylinder indicated schematically at the lower left of *Fig.* 3, and identified in *Fig.* 2 as "follower cylinder".

Reference to the illustration of the workpiece at the lower part of *Fig.* 1 shows that the slot profile has a gradual slope followed by a steep section corresponding to the feathering of the propeller blade. While the profile of the master cam provides the proper relation between the reciprocation and rotation of the work spindle, it is necessary to slow down the rate of movement when the steep portion is encountered. How this is accomplished is best illustrated in *Fig.* 3. The two cylinders which actuate the work fixture are alternately supplied with oil from a controlled-shift valve which, in turn, is operated from a pilot valve that is tripped by dogs attached to one of the plungers. Oil discharge from the returning cylinder passes through one or both of two meter valves which control plunger speed. The second meter valve may be bypassed through a cam-operated valve which

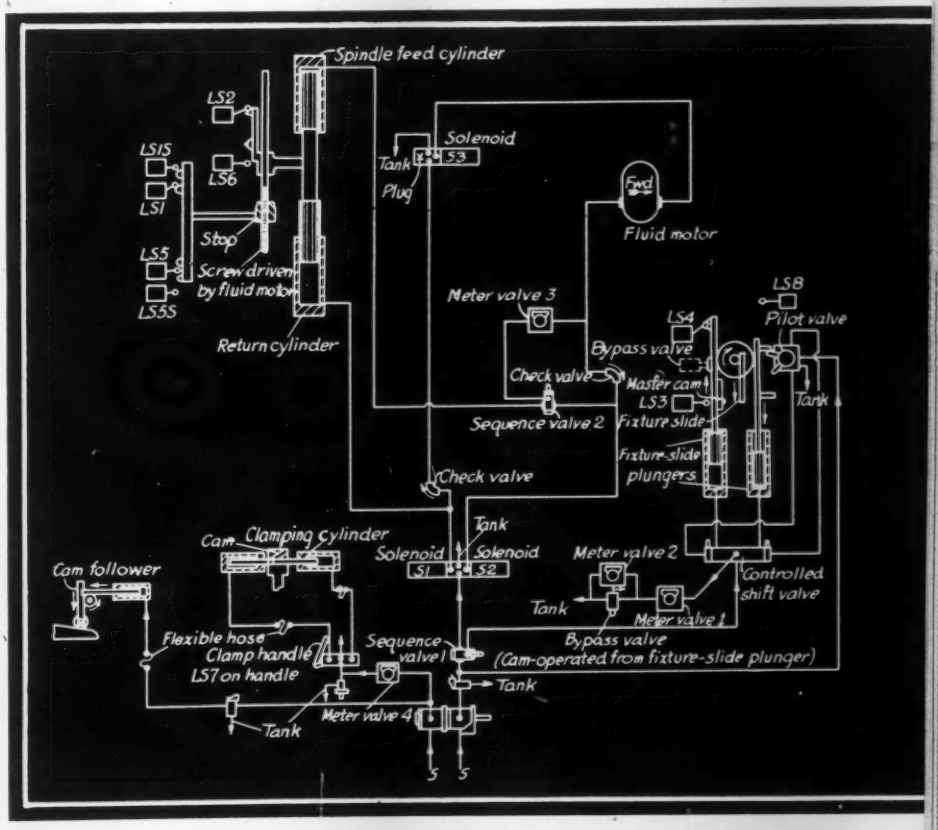


Fig. 3—Schematic diagram shows features of hydraulic circuit and location of limit switches, pilot valve and sequence valve controlling the cycle

is held open for high work-fixture speed and closed while the steep portion of the cam is being milled. This valve is opened and closed by a cam attached to one of the plungers.

Each spindle is powered by a 3-horsepower 1150-revolutions per minute motor driving through a double V-belt and worm drive, giving a cutter speed of 192 revolutions per minute, *Fig.* 4. The wormwheel is splined to the cutter spindle to permit the infeed of the cutter. Each motor is mounted on an adjustable base.

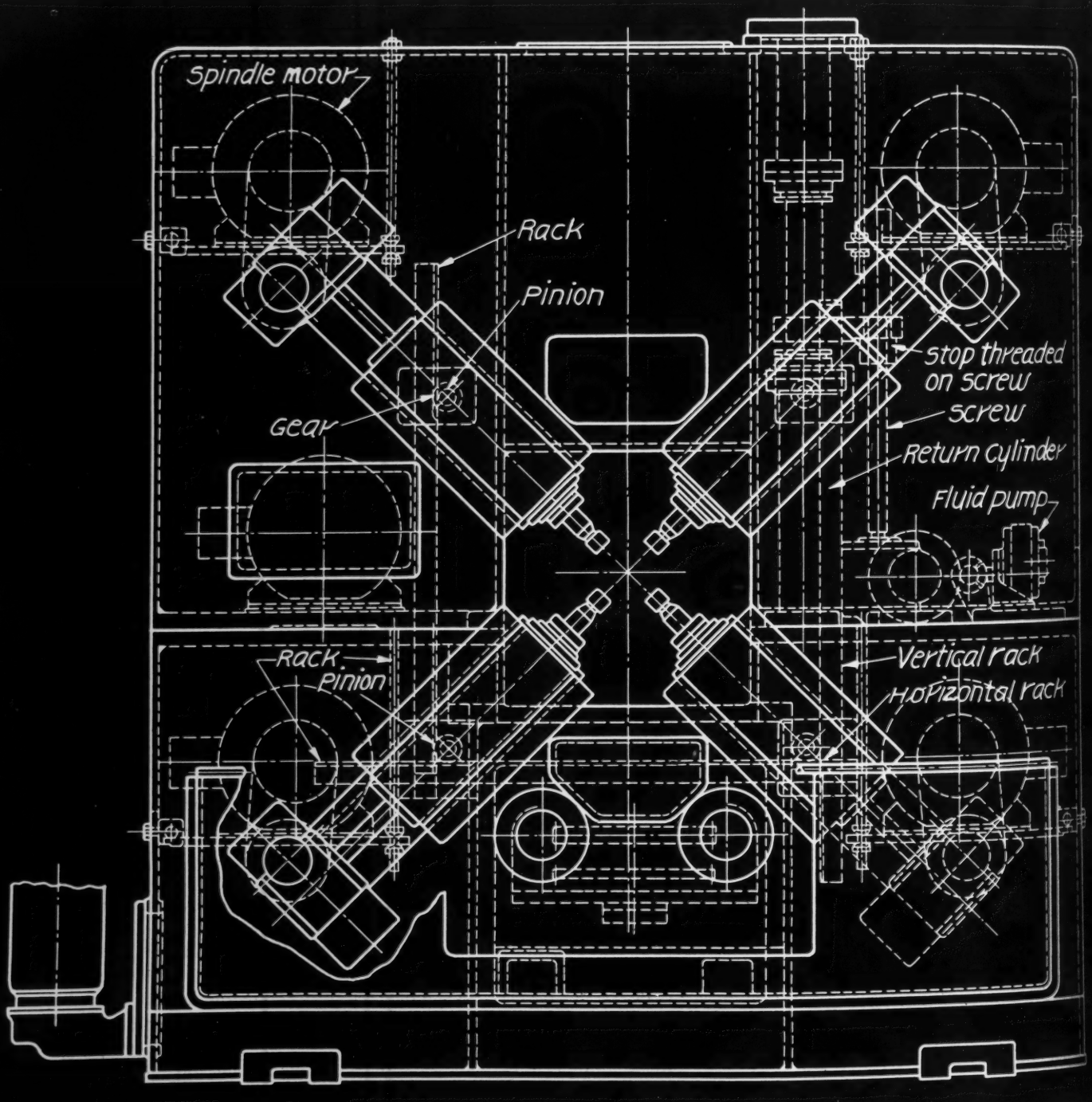
Infeed of the cutter spindles is effected by sliding the entire spindle housing, each housing being provided with a rack engaging a pinion as indicated in *Fig.* 4. On the same shaft as each pinion is a gear operated by another rack. Mechanical interconnection of the operating racks for each

spindle housing insures exactly the same feeding rate for each spindle. The mechanical arrangement is shown in *Figs.* 4 and 5.

The rack which operates the two right-hand units, *Fig.* 4, is attached to hydraulic plungers which also are indicated schematically in the upper left of *Fig.* 3 and in *Fig.* 5. The other two spindles are operated by auxiliary racks, one horizontal and one vertical, which transfer the motion from the pinion of the lower right unit to the lower left unit and then to the upper left unit, *Fig.* 4.

Infeed of the spindles consists of the following movements: A rapid advance up to the work, then short feed in small increments, one increment for each stroke, then quick return at the end of the cycle. Control of the movements is effected by a stop threaded on a screw, *Fig.* 5, also indicated at the upper left of *Fig.* 3. This stop prevents inward feed beyond a desired point at any time. Position of the stop is controlled by a hydraulic motor which drives the vertical screw through a bevel,

Fig. 4—Below—Cutter spindles are fed hydraulically, feed being controlled by stop actuated by fluid motor



spur and worm-reduction gear train.

Control of the feed movement may best be understood by reference to the hydraulic circuit diagram, *Fig.* 3, and the electrical circuit diagram, *Fig* 6. Shown at the lower center of *Fig.* 3 is a solenoid-operated spool valve which admits oil to either the upper or lower spindle feed cylinders for feeding and return. When the cycle pushbutton, *Fig.* 6, is pressed, the solenoid *S*1 is energized and oil flows through sequence valve 2 to the spindle feed cylinder. Spindle head advances and when it comes against the stop, flow to the cylinder ceases. Fluid spills over at sequence valve 1, indicated immediately below *S*1 and *S*2, actuating the fixture table. When the fixture table reaches either end of its stroke, a limit switch *LS*3 or *LS*4 is tripped, which in turn operates a time-delay switch and control relay which energizes solenoid *S*3.

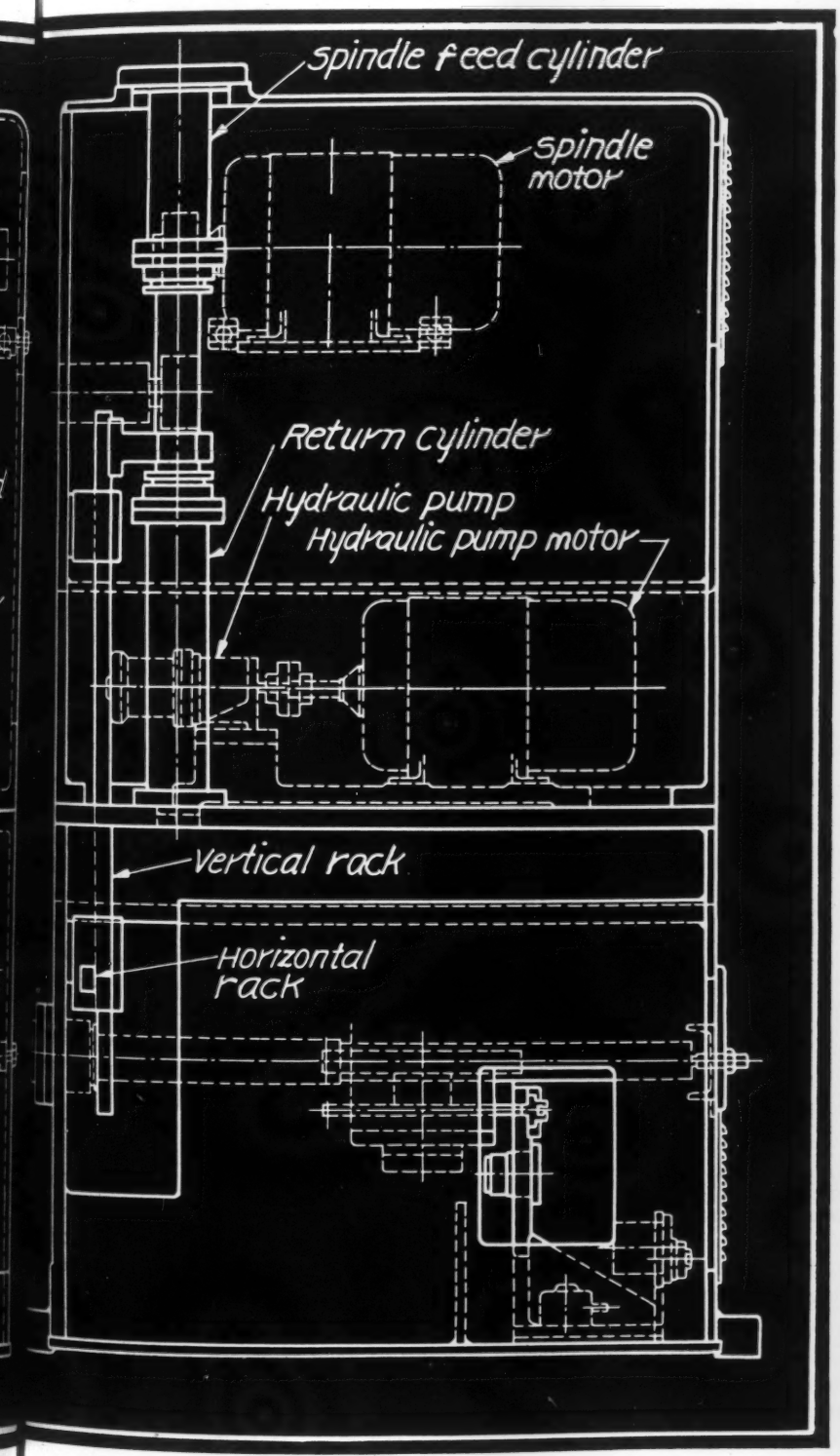


Fig. 5—Left—Side elevation shows hydraulic cylinders and racks for cutter spindle feeding mechanism

The valve associated with *S*3 normally is held by a spring in such a position that the outlet from the fluid motor is blocked by a check valve. However, when the solenoid *S*3 is energized, the fluid motor discharge is released to the tank, permitting the motor to rotate. As previously mentioned, the fluid motor actuates the screw which carries the spindle feed stop. Fluid flow to the motor passes through sequence valve 2 and meter valve 3. Since the fluid is metered, the motor turns briefly, rotating the screw, lowering the nut slightly and permitting the spindle to feed in a small amount which is adjustable within the limits of .01 and .15-inch. This occurs at each end of the stroke of the work fixture.

Retracting the Cutters

When the spindle has been fed to full depth, the limit switch *LS*5, operated by the stop, energizes solenoid *S*2 and *S*3 and de-energizes *S*1. Fluid is now supplied to the return cylinder and the spindle heads move outward. When the spindle head reaches the outer end of its stroke, limit switch *LS*2 is tripped, de-energizing solenoid *S*3 and permitting fluid to flow through the check valve and to operate the fluid motor in the reverse direction, resetting the spindle feed stop. To insure that the spindle head shall return only at the end of the stroke of the fixture slide, the limit switch *LS*8 is provided, this being placed in series with the limit switch *LS*5. The drop in pressure due to the rapid flow of oil to the return cylinder allows the sequence valve 1 to close, shutting off oil supply to the fixture cylinders. As the stop returns to its initial position, the limit switch *LS*1 is tripped and the solenoid *S*2 is de-energized, allowing its associated valve spool to be centralized by the spring.

Fluid pressure for the hydraulic circuit is furnished by a vane-type pump

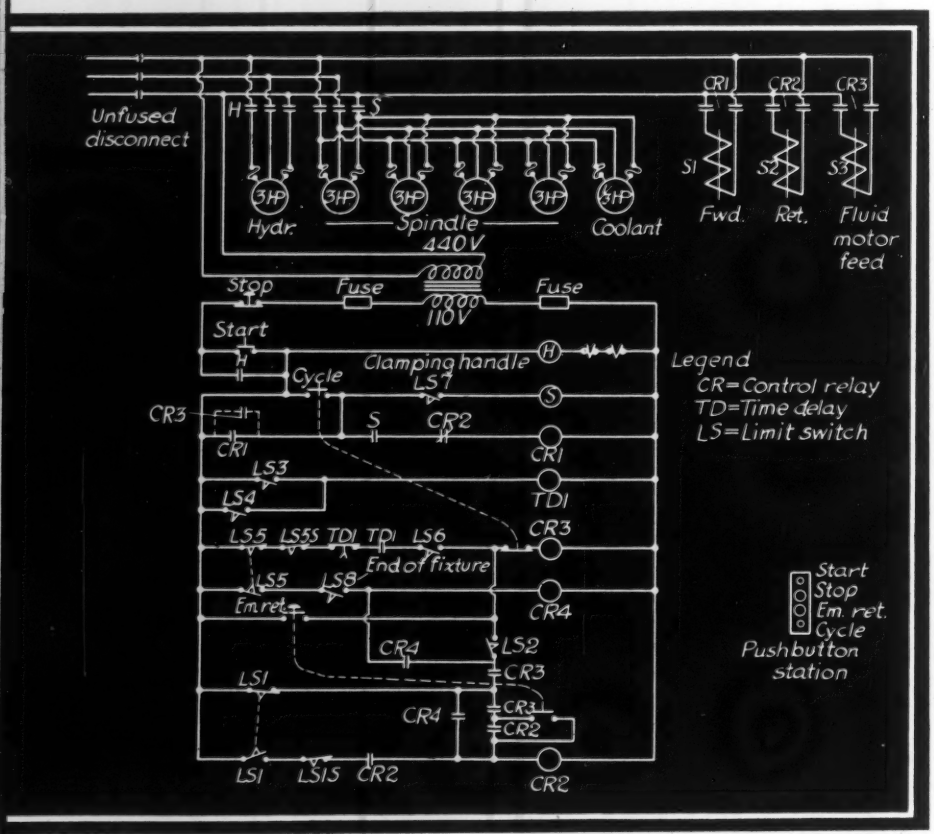
driven by a 3-horsepower motor. The feed control meters, which are adjustable, are placed at a central location and may be reached by removing a panel at the back of the machine, *Fig.* 5, which also encloses the other hydraulic valves. The oil reservoir or tank, which has a capacity of 47 gallons, also is located in the base of the machine adjacent.

Coolant reservoir extends from the open end in the front of the machine, *Fig.* 1, into the interior. The impeller-type coolant pump and its motor are visible at the lower left of the photograph. Also visible in the photograph is the shut-off valve which stops the flow of coolant to the spindles when the cutters are at the end of their stroke. This valve is operated from the upper left-hand spindle head by a cam which engages a roller on the end of a lever attached to the valve.

Adjustments Made Only by Authorized Person

Frame of the machine is of all-welded construction and provides complete enclosure for the drive and control mechanisms. Adjustments necessary in setting up the machine can be made only by an authorized person after removal of a bolted-on cover plate. Tool spindles, fixture spindle and the gears are heat-treated chrome-vanadium steel, while the wormwheels are hard tractor bronze. Main bearings for the work fixture spindle and also for the cutter spindles are of the tapered roller type.

Fig. 6—Below—Electrical circuit controls solenoid-operated valves in hydraulic system, sequence being completely automatic



It has been found that the spline mill principle of feeding the cutters into the work step-by-step results in excellent finish and good tool life. While the machine as discussed is used for finish-milling the slots, it may be adapted to a rough milling operation in which the step-by-step infeed of the cutters would be eliminated, the slot being milled in one cut, leaving from one-sixteenth to one-eighth of an inch of stock in the slot for the final finishing operation.

Diesel Engines for Naval Drives

PRINCIPAL characteristics of diesel engines for the Navy are light weight, compactness, and reliability of performance even when handicapped by being a long way from home. Justification of these requirements can easily be visualized in the light of everyday war news and these are the requirements which will help American engines to survive the competition of the postwar world.

Of striking significance is the fact that 47 per cent of all diesel engines in the Navy are of 4¼-inch bore, developing 33.3 horsepower per piston at 2100 revolutions per minute when operating on a 2-stroke cycle, and 25 horsepower at 2600 revolutions per minute when of the 4-cycle type. Overwhelmingly preponderant are the 2-cycle engines and engines of small bore, while the most widely used 4-cycle engines are of the supercharged type. Only the engines for standard ships' boats are naturally aspirated, and this is only because the desirability of freezing the design of these engines as originally adopted was far greater than any advantages to be derived from modernizing them.

Effect of Mass Production

The diesel engine never reached its maximum usefulness until adopted by the automotive industry under pressure of war and produced on assembly lines with greatly increased speed and with consequent reduction in price. Adaptation of the diesel to mass production has robbed it of the aura of craftsmanship which surrounded the process of assembling and testing a diesel engine only a few years ago. But it gained immensely in adaptability, as illustrated by the 4¼-inch 2-cycle cylinder which is used in the Navy in engines from 16 to 800 horsepower in units of from one to 24 cylinders. Thus the advantage of mass production forced the development of units having greater numbers of cylinders, since only by so doing could the Navy obtain the desired ranges of engine output.—*From a paper by E. C. Magdeburger, head engineer, bureau of ships, Navy department, presented at the recent war engineering annual meeting of the S.A.E. in Detroit.*

Design Pointers in Cable Control Systems

By R. R. Richolt
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Lockheed Aircraft Corp.

CONTROL systems for aircraft should be designed on the basis of deflection rather than strength. This has been demonstrated on every new ship built where the parts—such as pulley brackets, bell cranks and cables—are strong enough but do not have sufficient rigidity. The resulting deflection always prevents the application of the design loads used and at the same time gives the pilot a "springy" control.

Increasing the design loads to be used is no remedy for this condition. It causes the addition of weight where it is really unnecessary and still does not prevent

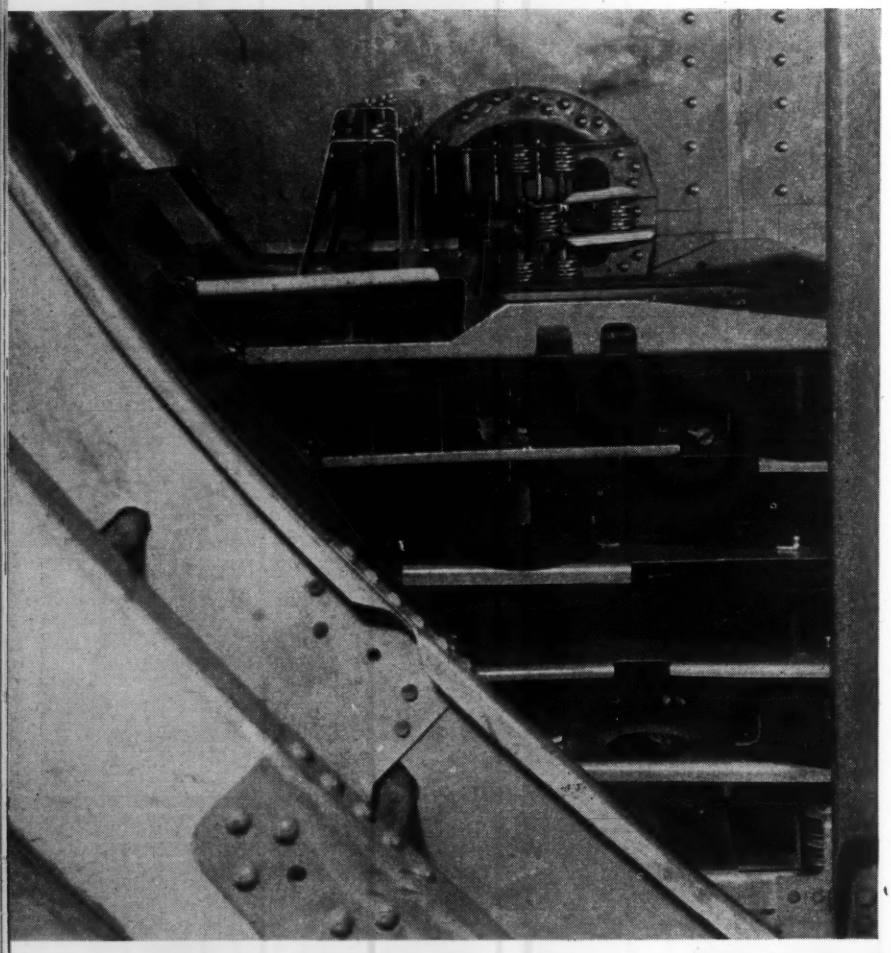


Fig. 1—Above—Control pulley bracket for power plant controls

deflection troubles. As an example, a push rod used as a column will not gain much in rigidity by increasing its design load and requiring large bearings and large diameter tubing. The larger design load will help in obtaining more rigid pulley brackets and supporting structure but, if some deflection method of analysis could be used, the pulley brackets would be designed for rigidity and the push rod could be made a reasonable size. These are only two examples, but they demonstrate the fallacy in a large percentage of the control system design.

Cable size is always determined by the *EA* value and the length, if appreciably long. Standard 7 x 19 extra-flexible aircraft cable is poor in this respect. It stretches considerably in comparison with 1 x 19 cable or similar cable, but was always required in the past because of the bends necessary. If the crooked path through which the cable is routed to clear such things as nose wheel wells, baggage compartments, etc., could be eliminated, 1 x 19 cable would be entirely satisfactory.

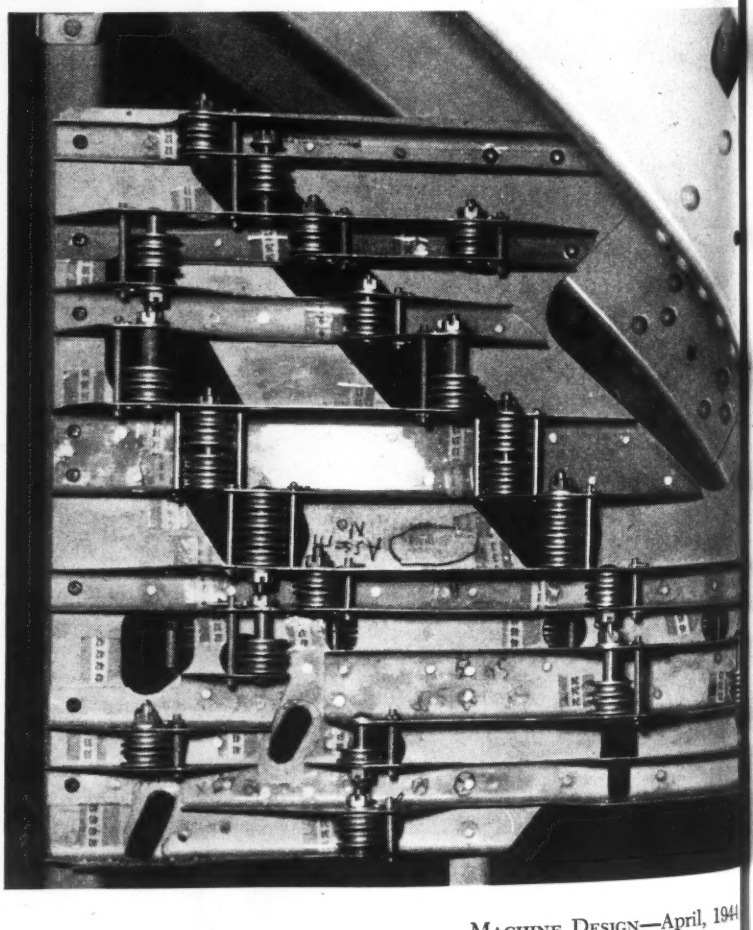
Shown in *Fig.* 1 are typical control pulley brackets for power plant controls. Cables have not been installed but the number of pulleys clearly indicates the complication of a cable system of this sort. In *Fig* 2 are brackets for roller fairleads which replace the usual phenolic strips or blocks to reduce the cable system friction. They are nothing more than small pulleys and are used for small angles of bend on the cable.

Straight Tunnel Would Be Ideal

The ideal control would utilize a straight clear tunnel or space through the fuselage to the tail and through the wings to the ailerons so that pulleys could be eliminated entirely. In fact, with airplanes over 100,000 pounds gross, this may become a necessity not only to reduce the stretch in the cable system, but also to keep the friction of the system within reason.

If, on a large airplane, space could be allotted through the length of the ship for this purpose, cables could be dispensed with entirely and dural tension rods used. These rods would be supported on swinging cranks at the ends and at intermediate points. This would then cause pin-joint friction only instead of the cable friction as we now have it.

Fig. 2—Below—Roller fairleads are employed instead of phenolic blocks to reduce friction and provide for small angles of bend on cable



Contrary to what many people believe, the greatest amount of friction in a cable system is caused not by the pulley bearings and other linkage bearings but by the internal friction of the cable itself in bending over the pulleys.

A dural tie rod system would have three other advantages: First, it would require a minimum service time because of the fewer parts; second, the reduction in weight for the same reason; third, insofar as the smaller planes are concerned, it would eliminate the necessity for cable tension compensators to take care of differences due to thermal expansion. Elimination of hundreds of drawings for pulley brackets alone would be advantage enough from the engineering time standpoint to make this worthwhile. Cost in manufacturing would be reduced by the same proportion.

With respect to the third point mentioned in the previous paragraph, temperature in larger planes is only one of the causes of change in rigging tension. Deflections such as are caused by pressurizing the cabin actually change the lengths of the fuselage slightly. Also, temperatures are not consistent in all parts of the ship. Control cables or rods may be at one temperature and the skin of the ship at another. Therefore the larger the airplane the more the necessity for some means of maintaining the rigging tension at a constant value to reduce friction.

A compromise between dural tension rods and steel is used by Lockheed. This is Alclad cable, consisting of the regular extra-flex cable which is stiffened by aluminum alloy tubing swaged over the outside. The swaging increases the *EA* value tremendously. Where the cable must pass over an idler pulley the aluminum alloy tubing does not cover the cable for a short space.

If a tunnel for this purpose is planned in a new airplane, it should be outside the pressure area of the cabin. This might mean that it would take the form of a keel or fin along the belly of the fuselage.

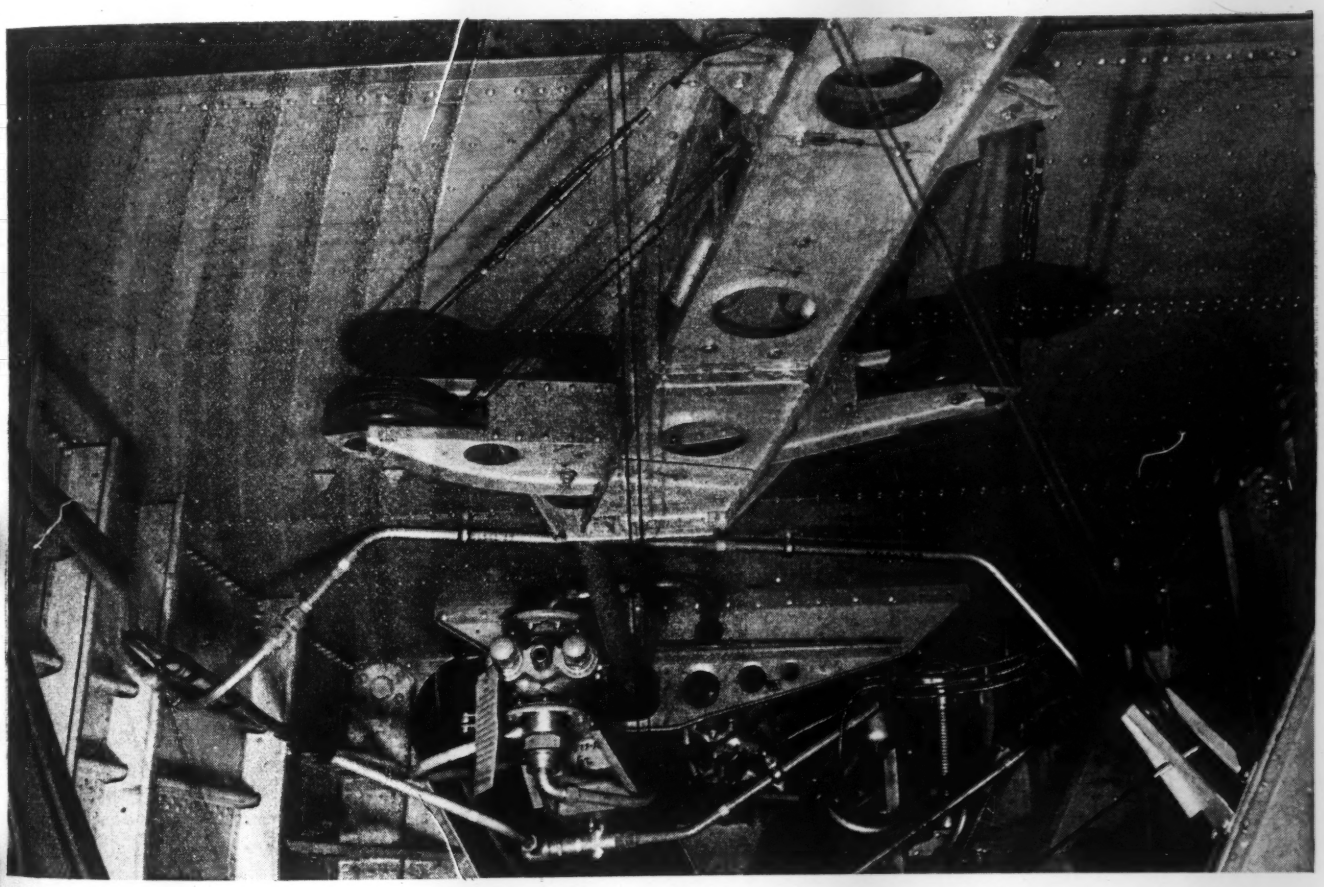


Fig. 3—Block and tackle principle is utilized to increase cable travel. Bracket supports cable-tension regulator

Hinged doors for maintenance and inspection should be located at strategic spots along this tunnel. They should be large enough to *work* through instead of just *peek* through. Making the maintenance easier will mean that the control system is really serviced instead of passed over because it is too hard to get at. Manufacturing time would be reduced for the same reasons.

A method of reducing control system deflection is to reduce the load in the cables by increasing the cable travel. This method is used on Lockheed's Constellation. Space restrictions, however, do not allow use of a simpler lever. Increasing the length of the lever to which the cable is attached would be the most desirable and most efficient from a friction standpoint. On the Constellation, a block and tackle arrangement was finally decided on as the best

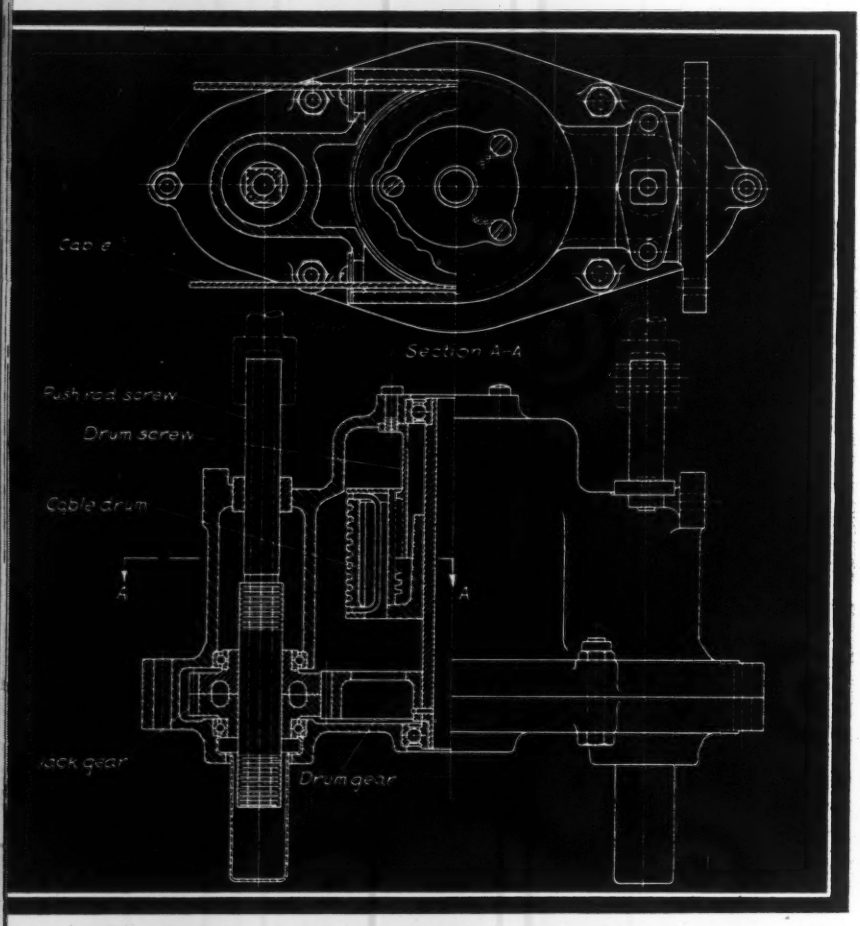


Fig. 4—Tab control jack. Unit has low efficiency at low loads

arrangement that would go into the space available. The additional pulleys involved penalize the system because of the friction added. *Fig.* 3 illustrates a block and tackle application for increasing cable travel. This particular bracket is built to support the cable tension regulator. The "dead ends" of the cables are attached to the regulator unit which maintains a constant rigging tension on the cables.

Alternative Systems Include Gearbox

It may be possible to improve this system by the use of a gearbox to replace these pulleys. This gearbox would be required to generate the 22¼-inch cable travel with the full angular throw of the control column, rudder pedals or control wheel, and would be required to do this with at least the equivalent or better efficiency than the cable method used. The gearbox might have a drum for winding the cable or might use a straight rack or chain. Another possibility might be a hydraulic transmission where the stick operated a large diameter piston which, in turn, operated a smaller piston (or pistons) with a long stroke. In the latter case the unit should be complete with its own oil reservoir built into the unit. Probably the type of system which would be the most acceptable would use a mechanical gear train driving a sprocket. The sprocket would drive either roller chain or silent chain spliced into the control cable. The chain would have less friction than a cable wrapped on a drum.

A problem exists in the multiplier for the control wheel on all aircraft. In this case the minimum diameter of sprocket is used at the wheel shaft to reduce the interference of vision of the instrument panel as much as possible. This means that the chain or cable in the column has a short stroke. This short travel then must be multiplied preferably with the same type of gearbox as used for the elevator and rudder controls, although the driving mechanism is different. The reduction of cable load is a necessity because the aileron system is generally the worst for deflection.

There has been a great variety of tab control jacks and operating units designed at Lockheed. All have some good characteristics and some bad ones. The one used on the Constellation had an additional requirement—two push rods plus an extremely high load.

The high design load made it necessary to reduce the number of turns of the pilot's control—decreasing the mechanical advantage—in order to use 3/32 cable for the drive. This restriction plus the double jackscrews, the long cable run and the number of jack units (three on the rudders, two on the elevator, and two on the aileron) all combine to make an inefficient system that requires a fairly high torque at the cockpit unit to operate the tab if under any appreciable load. The efficiency of these jack units has been measured at approximately 20 per cent maximum. At low temperatures, the efficiency is still lower if an attempt is made to operate the unit at normal speed, due to the grease.

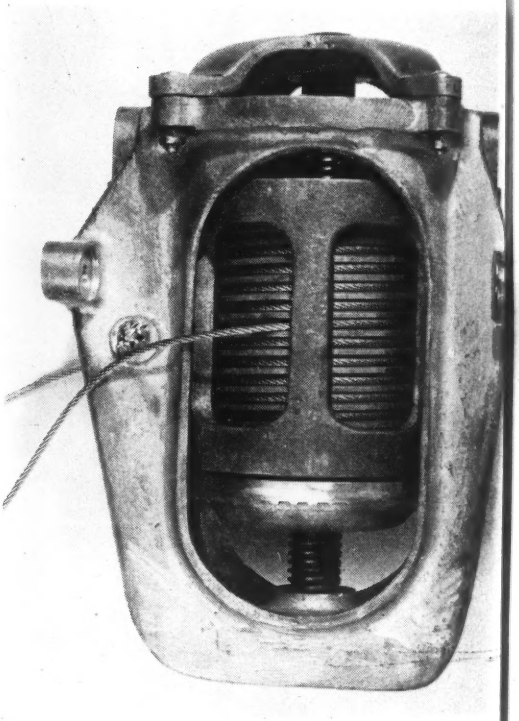


Fig. 5—Redesigned tab jack having an average efficiency of 35 per cent which goes as high as 60 per cent at low loads

Most of the friction in the tab operating system is in the Acme thread screws of these

units. If there were some other dependable mechanical device which would do the job with less power loss, the complete system would be improved greatly, especially from the pilot's standpoint. On larger airplanes where the cable run would be still longer and the loads on the push rods higher, it would be necessary to pay as much attention to friction losses in the tab control system as in the main flight controls.

Jack Efficiency Is Low

The previously mentioned tab control jack is shown in *Fig.* 4. Its undesirable characteristic of 10 to 12 per cent low-load efficiency and a maximum of 20 to 25 per cent at high loads is the direct opposite of what is most desired since the operation in the ship is practically all at the low-load condition. The design illustrated in *Fig.* 5 was built to get around this difficulty. Tests indicate an average efficiency of about 35 per cent although at low loads it goes as high as 60. This means it would be reversible if it were not for the friction in the cable system. Due to this friction in the cable driving the unit, it is irreversible as desired but operates easily for the trimming condition in flight. The main reason for this higher efficiency is in the Acme thread angle used. Since the efficiency depends upon the tangent of the helix angle of the thread, as well as the coefficient of friction, an attempt was made in this design to make this value as low as possible without reversibility. The curve in *Fig.* 6 illustrates this and laboratory tests conducted on the unit proved the theory. The ratio of cable travel to push rod travel is 36.8 to 1. This gives a 72-inch cable travel for 1.96-inch travel of the push rod. Both the tab units shown in *Figs.* 5 and 6 were designed for the same loads and both were built to operate dual push rods.

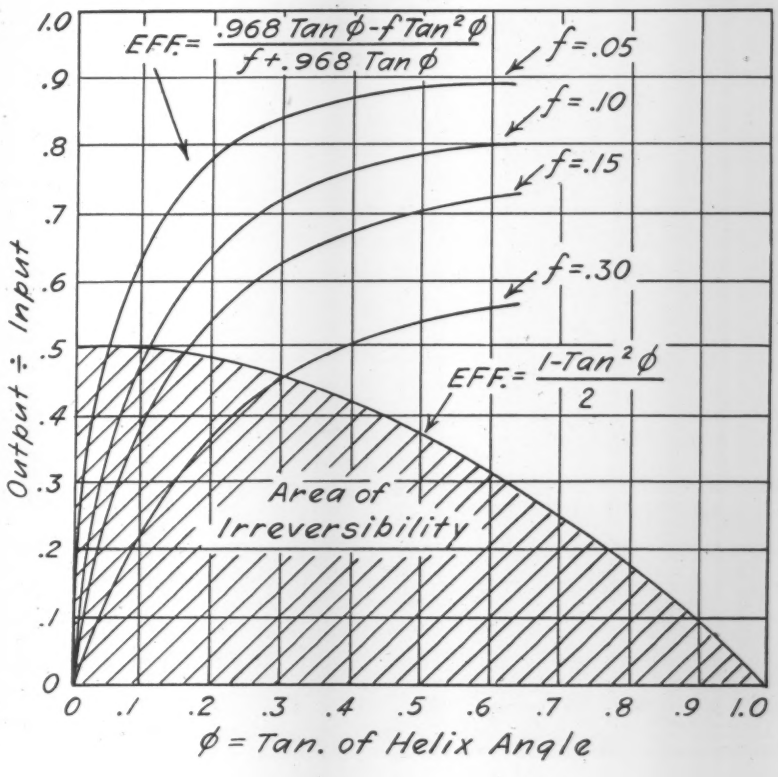
A source of constant trouble on production airplanes is the wobble and lack of clearance on control pulleys. Pulleys in sheet metal pulley brackets rub the bracket because the holes in the bracket assembly were not drilled normal to the plane of the bracket or because the bearing in the pulley is loose and allows too much wobble. AN specifications limit the wobble clearance in terms of thousandths of an inch, but aircraft shops do not build sheet metal brackets with this kind of accuracy. The result in the complete assembly is cause for many rejection tags.

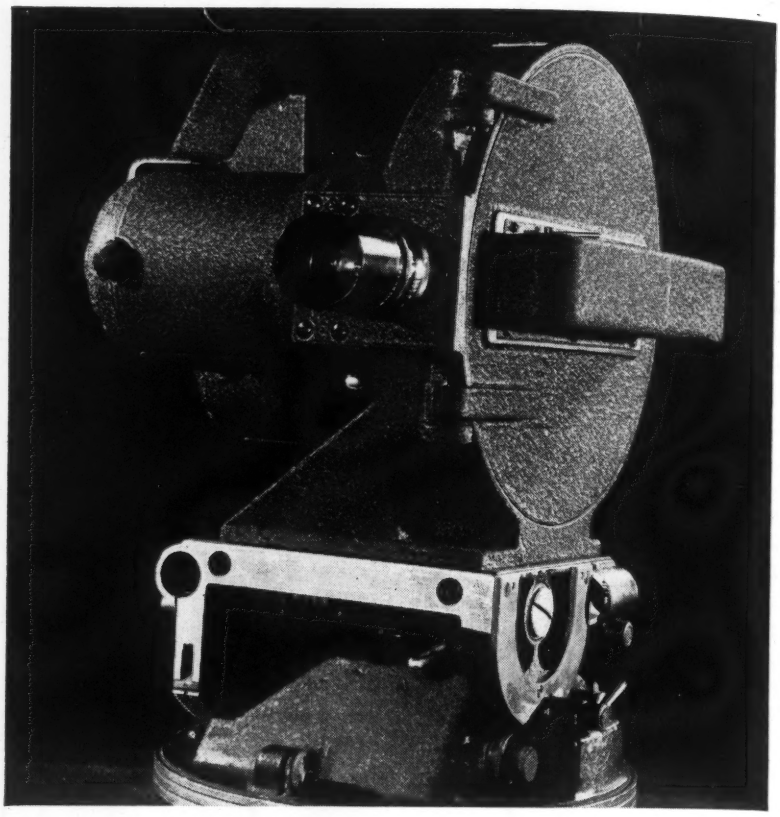
There seems to be no chance of improving this condition until after the war because the basic trouble is in the design of the pulleys and, obviously, standard parts could not be altered at present. However, a set of dimensions should be arranged for a new series of pulleys which would eliminate rejection expense in the future. All aircraft manufacturers are having the same trouble so the standard would be accepted as soon as it could be set up.

In this new series, the dimensions should be determined with an eye to installation in an airplane instead of looking at the pulley alone as an isolated part. Bearings used in the pulleys should have a large bore to allow a number of pulleys on the same bolt without extra supports on the bolt to relieve bending stress. Minimum number of different bore sizes is desirable to allow stacking different diameter pulleys on the same bolt where necessary. The hub projection on the bearings should be generous so that it would be unnecessary to utilize close fitting bearings to prevent rubbing of the pulley against the bracket or adjacent pulleys on the same bolt. This, incidentally, would make the pulley cheaper to build since "conrad" type bearings could be used with plenty of capacity because of the larger bore diameter, and the installation tolerances of the bearing in the pulley would not be so critical. The overall length of the bearing hub should not only be generous for the pulley itself, but should be correlated with the pulley pitch diameters so that double right-angle bends could be made with control cables, as in the P-38. There are a number of other considerations which would become apparent upon investigation.

In order to achieve the ultimate in pulley design, some device should be incorporated in the pulley bearing which would allow the pulley to be aligned with the cable and then clamped. This must be accomplished with the minimum in weight, space and complication. If the cables could be threaded through the various brackets, the pulleys aligned and then clamped in position, the fabrication and installation problem in control systems would become much cheaper and easier.

Fig. 6—Efficiency curve for redesigned jack indicating area of irreversibility in the shaded section beneath the curve





Ultraspeed Camera Utilizes Constant Feed

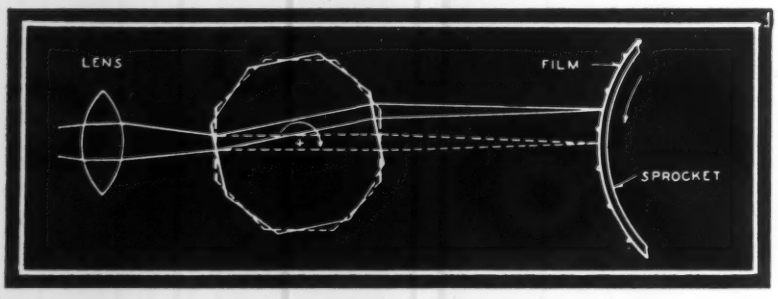
HIGH-SPEED cameras, utilizing intermittent film drive and mechanical shutter, present difficult design problems. It would be almost impossible, for ultra high-speed work, to keep the film motionless during the exposure and the shutter synchronized perfectly throughout the speed ranges of the camera. Stresses on the drive mechanism and film would increase tremendously with speed because of the intermittent drive. Furthermore, the exposure time permissible with a mechanical shutter would be of such short duration as to require extreme light intensity on the object being photographed.

Exposures up to 9000 frames per second are possible, with the camera illustrated, by utilizing a constant-speed drive for the film and an optical system to replace the mechanical shutter. Two universal motors are used: One, having adjustable speed, drives the film sprocket and optical "shutter"; the other provides a constant tension on the take-up reel. Electrical characteristics of this second motor allow for the necessary slip to compensate for the increasing diameter of the reel during take-up, obviating a mechanical clutch or slip arrangement.

Shutter for this camera, developed and designed by Bell Telephone Laboratories Inc., employs a rotating compensating prism. As the prism rotates, the image formed by the lens at the film plane moves in synchronism with the film to produce a sharp image. Thus, as the film passes from the feed spool, under a hold-down roller and onto the sprockets, is stripped off the sprocket and fed to the take-up reel, the image follows each frame during the exposure interval as indicated in the schematic diagram.

Synchronization of the prism and sprocket is obtained through a five-to-one gear ratio and may be adjusted readily after assembly of the camera. Sprocket wheel necessarily is extremely accurate, with the sprocket teeth slightly larger than the perforations in the film. This aids the film to ride over the sprocket without vibration or shaking. Fresh, properly humidified film must be used so that film shrinkage will not cause binding. Pictures can be taken at speeds up to 9000 frames a second for an 8-millimeter model and 4000 for a 16-millimeter camera.

Schematic of optical system utilizing eight-sided compensating prism which allows film to be driven at constant speed



SOLENOIDS— For Power For Control

By M. L. Jeffrey
Manager, Electrical Mfg. Div.
National Acme Company

WHERE quick, remotely controlled mechanical motion is required, the machine designer would be hard pressed to find a more desirable power unit than the solenoid. It is compact in size and relatively light in weight; supplies power remotely located from control without the complicity of other mechanical parts and units; is extremely simple in design and rugged in construction, insuring long, trouble-free service life; can be readily wired into electric power supply—available for primary

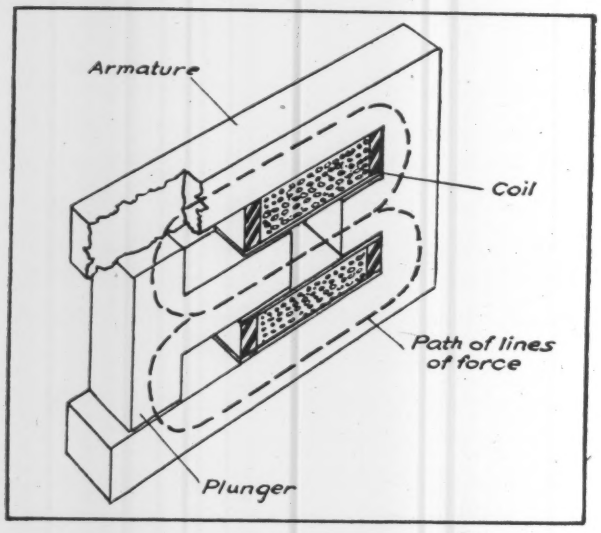


Fig. 1—Elements of the solenoid. Note the maximum continuity of iron path provided for magnetic flux

drives of practically all mechanical equipment—without any special knowledge of electricity or electronics.

An outstanding example of the dependability of properly engineered solenoids is their application in the firing controls of the twin Bofors antiaircraft guns shown in the official U. S. Navy photograph which heads this article. Actually, there are a number of solenoids used in the vital mechanisms of these gun controls.

In dealing with the proper application of these handy power units, it probably will be well first to present a brief resume of their operating principles.

Every solenoid consists of three essential units; coil, armature and plunger, in general arranged as shown in *Fig.* 1. When an electric current passes through the coil, it inductively sets up a field of magnetic lines of force, the magnetic attraction of which pulls the plunger into the coil. It is a recognized fact that air offers more reluctance to the concentration of magnetic lines of force than does iron and, since the magnetic pull of the coil is dependent on the number of lines of force in the field, to concentrate as many lines as possible around the coil an iron path is provided by the armature in combination with the plunger.

Pull Varies With Stroke Position

It will readily be seen that any break of continuity in the iron path provided for the lines of force will result in an increase in reluctance for the overall magnetic circuit and thus diminish the effective pulling force of the coil on the plunger. This is the reason why, in all solenoids, the pull exerted by the coil decreases as the plunger is moved to its outermost position. In other words, the power of all solenoids is less at the beginning of their stroke than when they are fully seated with their plunger end in contact with the surface of the armature inside the coil.

In specifying a solenoid for a particular application, the designer must be sure to take into account this variation in magnitude of pull from one end of the stroke to the other, otherwise he may find the solenoid specified has insufficient power for the job.

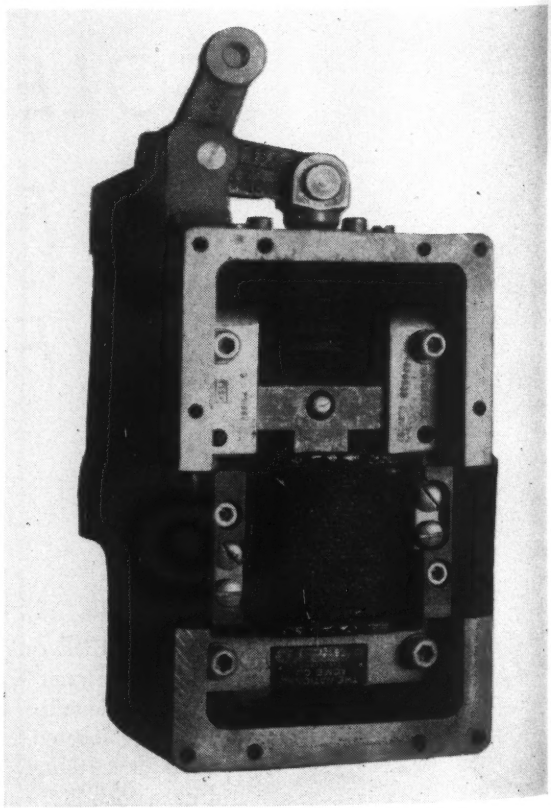
An interesting point—at first seemingly paradoxical—in regard to the immediate foregoing, is the fact that amperage is inversely proportional to the rate of change in pull. In other words, when the plunger is fully seated within the coil, exerting its maximum pull, the amount of current drawn by the coil is less than when the plunger is at its outermost position.

Hard-Facing Increases Life

It has been stated that solenoids, being inherently simple in design and rugged in construction, have long trouble-free life. Nevertheless, extensive research work is bringing to light improvements in solenoids of such nature as to insure greater dependability than has been possible in the past. One of these improvements, lately patented, is incorporated in the solenoids used in the control mechanisms of the antiaircraft guns illustrated. It is partial hard-facing of the armature and plunger contacting surfaces. There is sound engineering logic behind this hard-facing procedure. Too often a solenoid, after functioning efficiently over a period covering thousands of operational cycles, will commence to behave in a disconcertingly erratic manner. Once this erratic behavior starts, it rapidly becomes worse until the solenoid ceases to function.

What has happened? More often than not, through the incessant pounding of plunger core piece on armature seat, the plunger end has "mushroomed" in much the same manner a cold chisel end does under the action of many hammer blows. Thus the plunger binds in the armature with the result mentioned.

Fig. 2—One way to make a solenoid unit watertight. Case-to-plunger seal is rubber bellows-type sleeve



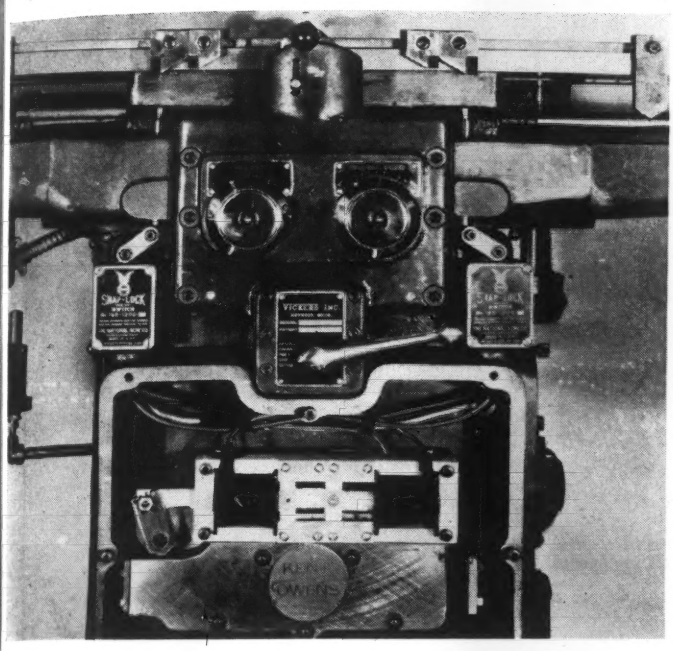
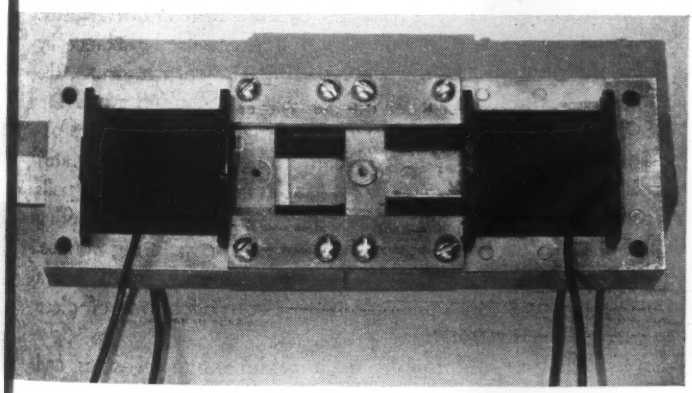


Fig. 3—Above—How a pair of solenoids can be mounted in line. Here they actuate the work-table control of a milling machine

Fig. 4—Below—Closeup of above unit, showing a typical method of designing two solenoids into an effective "push-pull" device



However, by providing the plunger end as well as the armature seat with a partial hard surface—in this particular instance a welded deposit—the mushrooming is effectively curtailed and the service life of the unit substantially increased.

For applications which require that the solenoid be mounted at a location where it will be exposed to weather, splash or actual immersion, it becomes necessary to enclose the solenoid in a suitable watertight case. Designing such a case usually is a simple matter. However, sealing the plunger connecting rod can be somewhat more difficult. A practical solution to this problem is shown in *Fig.* 2, in which a bellows-type rubber sleeve effects a seal between the case and plunger. This particular assembly was designed to withstand a 2000-pound shock test such as is often prescribed by the government for solenoids to be used on board ships.

Wherever possible the connection between the solenoid plunger and the member it is to pull should be so designed as to permit the plunger a short pre-travel, say up to one-eighth of an inch, before it actually picks up the load. This procedure permits the plunger to establish some momentum before the working portion of the stroke and also decreases effectively the heating of the coil.

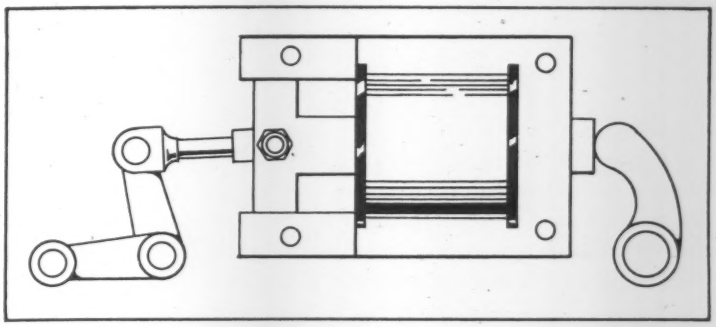
There are some instances where it is impracticable to achieve the desired mechanical motion by employment of a single solenoid. Such problems often can be solved by using two of the units as in *Fig.* 3 which shows an application of a pair of solenoids on a well-known milling machine where they control the forward and reverse traverse of the work table. A more detailed view of this solenoid unit is shown in *Fig.* 4 which serves to illustrate one of the methods for utilizing two solenoids for control of the same mechanism. In this case the solenoid plungers are linked together so that one solenoid pulls the lever and the other solenoid pushes it. They also may be mounted in tandem, in which case the total pull they exert is equal to the sum of the pulling forces exerted by each.

As with other types of power and control equipment used in the design of machinery, an intimate knowledge of the inherent merits and limitations peculiar to solenoids will go far toward helping the designer in putting them to work on a competent basis. For example, since the greatest amount of current is drawn by a solenoid coil when the plunger is in its outermost position, it readily will be appreciated that the unit's highest efficiency will be realized when the shortest possible stroke is employed. What actually happens when a short stroke is utilized is that the most favorable ratio of amperage and plunger force is established. Naturally, in addition to all this, the shorter the stroke, the more quickly it can be effected.

Make Sure Solenoid Is Strong Enough

It can safely be stated that in all but the most critical applications, the solenoid selected should have between 20 and 25 per cent more power than is theoretically required. This procedure is necessary because the variations in voltages found in most plants are such that the solenoid is required to

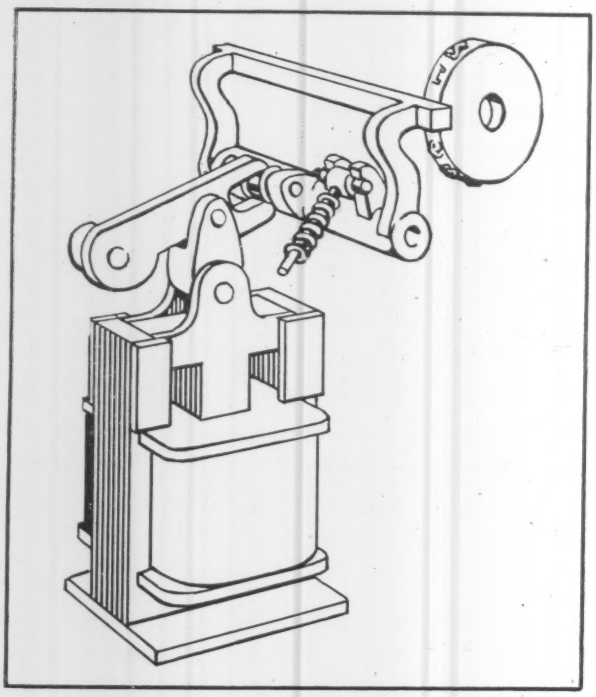
Fig. 5—"Through-plunger" type of solenoid permits energizing two mechanisms from single power source



function properly at a 10 or 15 per cent line drop in voltage and must be able to overcome additional loading that may develop from friction caused by sticky oil or dirty surfaces. Such loading is indeterminate and can be taken care of only by resorting to a safe margin of overpowering. Incidentally, the variations in line voltage might readily be on the plus side so that solenoid coils should be wound so as to withstand 10 to 15 per cent over their rated voltage.

It is always unwise to use a solenoid beyond the voltage range prescribed by its manufacturer. Not only does the solenoid function inefficiently in regard to power consumption, but its behavior becomes erratic and unreliable. Slight variations in the imposed loading on the plunger can, for example, result in improper seating, causing excessive amperage which in turn gives rise to overheating.

Possibilities in developing solenoid-actuated mechanisms are unlimited, being dependent only on the ingenuity of the individual designer. To help point the way toward a fuller comprehension of what can be done, representative mechanisms are pictured in *Figs.* 5, 6 and 7.



By using a plunger which passes right through the base of the armature, a solenoid is produced which can be used for either pushing or pulling or, as shown in *Fig.* 5, for both. Plainly, the mechanisms at each end of the solenoid may be accurately synchronized or, by permitting the pusher end a short free travel before actually contacting the lever, sequence operations can be effected. What is done with the motion imparted to these levers in subsequent interconnected mechanisms might be one or a few of many things. Hydraulic valves can be opened and closed, disk indexing can be effected,

Fig. 6—Right—Pull-down ratchet and pawl design as shown, for indexing counter dials, could also be adapted for controlled rotation of feed rollers

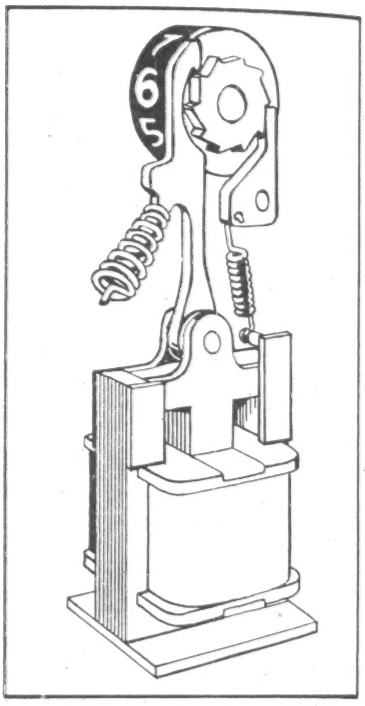


Fig. 7—Left—Automatic cocking and tripping of spring - loaded throw mechanism makes possible the quick, hammer-like action needed for the best kind of printing

clutches can be engaged and disengaged, etc.

An interesting ratchet-type mechanism is shown in *Fig.* 6. Here, counter dials are indexed step by step in accurate register. The hooked member is pulled down a prescribed distance by the solenoid plunger after which, during the return stroke, the ratchet wheel is held in position by the spring-loaded pawl.

For mechanisms employed in printing, best results are obtained by the use of a hammer-like action. Thus, as illustrated in *Fig.* 7, the solenoid is used so that when energized it pulls the hammer back and then, with the circuit broken, the spring quickly throws the hammer up to contact the printing surface. The mechanical action is such that the hammer is allowed a slight overtravel which permits it to drop back out of place after performing its function.

Solenoids are obtainable in a wide range of sizes and shapes and, while it usually is more economical to specify those which are stocked by a manufacturer, a special solenoid can be developed for applications in which the standard units would not meet the requirements. The specifications submitted to the solenoid manufacturer for development of such "specials" should be as specific and complete as possible. Maximum overall dimensions, shape desired, length of stroke, power expected of the solenoid, whether service will be continuous or intermittent and type of electric power available should all be set forth clearly and, where feasible, supplemented by a drawing or sketch showing the method planned for mounting.

Before drawing up the specifications for a solenoid, it is of course necessary to determine just how much force is required to do the particular job. Since the machine into which the solenoid must fit is not usually in actual existence, but merely drawn up on paper at the time of specification, the force needed to actuate that portion of the mechanism which will be powered by the solenoid should be carefully calculated. It is, however, wise to bear in mind that a number of the factors entering into such calculations are of a variable nature (for example, coefficient of friction). This being the case the solenoid specified should have more power than the calculations indicate.

Flexible Shafting Solves Remote Operation Problem

By George G. Eisenbeis
Chief Engineer
Stow Manufacturing Co.

POSITION of the various valves throughout a ship, some in inaccessible or remote places, makes necessary the installation of operating control stations at convenient locations—usually some centralized point. Centralization of controls is particularly important when, because of an emergency, speed in operating a valve may be vital. Control at the valve location by means of the regular handwheel also is desirable, except in the case of steam smothering valves which, of course, can be

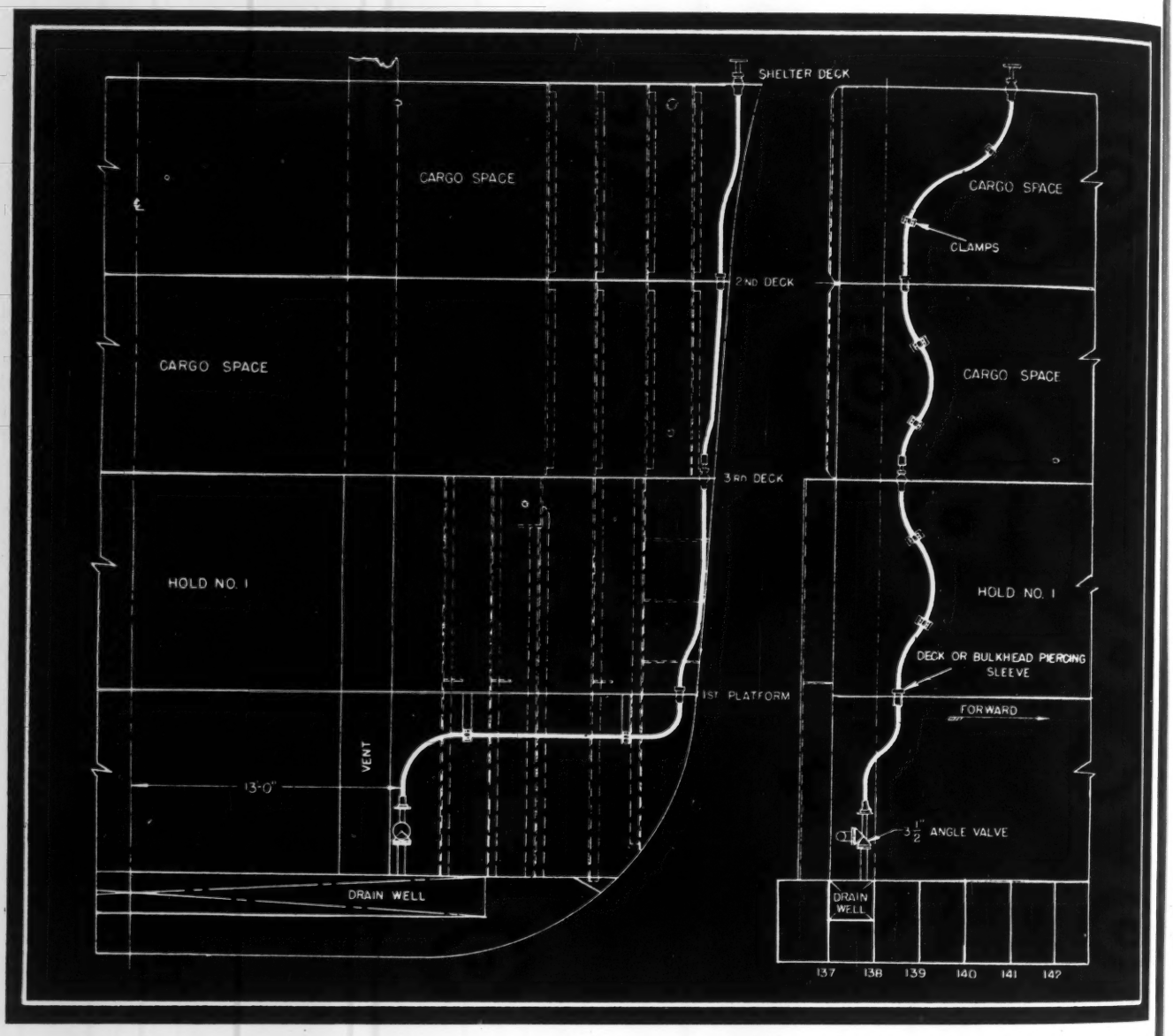


Fig. 1—Suction valve below decks is operated through flexible shafting from handwheel on shelter deck

operated only from remote stations.

Requirements of a mechanical drive for remote operation of valves on board ship may be understood from a consideration of the following factors:

1. Nature of the ship's structure as well as the location of other installations frequently requires the control shafting to be routed around corners.

2. Permanent distortion of the ship's structure can develop from a number of different causes, resulting in misalignment.

3. Relative movement of the ship's structure during operation induces a continually varying distortion.

Thus, when a ship is launched there is some settling and shifting of bulkheads and decks due to redistribution of supporting forces, the amount depending on the type of construction, settling of the ways, and other factors that are impossible to control. Between the light load and heavy load conditions distortion of varying extent will occur, although every attempt is made to control this by equal distribution of loads. Under war conditions heavy cargoes are the rule rather than the exception, and even in peacetime load lines may be disregarded on occasion, resulting in temporary or permanent distortion.

Even with the greatest care in loading, some shift of load may occur due to rolling and pitching in heavy seas, throwing additional stresses and strains on bulkhead and decks. Collision and near misses of bombs and shellfire cause distortion depending upon the heaviness of the impact or concussion as well as on the ship construction. Tankers or other ships with extra large holds suffer more severely than a compact, rigidly built vessel.

Apart from any shifting in cargo resulting from the motion of the ship, distortion is ever present in a ship underway, as evidenced by the creaking and groaning with which sea travelers are familiar. The overall length and beam of a ship largely controls this, particularly in vessels of latest design; some of the old collier-type ships have been known to work a foot and a half off their base line. While the ship is riding a wave amidships, strain is thrown on this point by the weight of the fore and aft portions of the vessel where the buoyancy is momentarily insufficient. An opposite effect occurs when conditions

are reversed and the fore and aft sections are riding a wave. This hog and sag cause the ship to rise and fall from its base line.

Until quite recently the standard type of control for remote operation of valves consisted of a series of rigid shafts connected by bevel gears, universal joints and ball-tooth joints. Such an arrangement was readily adaptable to routing around corners, but it was sensitive to any misalignment of supporting bearings, etc., and distortion of the types discussed in the foregoing frequently caused hard operation or even binding of the mechanism.

Increasing severity of the demands of wartime operation of valve control systems and the even greater necessity for reliable performance prompted the use of fully enclosed flexible shafting in new ships constructed under the emergency program, particularly the Victory ships which may use as many as thirty such shafts in each vessel. In addition to meeting the rigorous requirements as regards distortion, this type of control gives greater flexibility to ship construction schedules. While it is customary to follow a predetermined layout and to have a definite place in any new construction program for the installation of the flexible shafting, this schedule need not be rigidly followed. Should it be found necessary to make unforeseen structural changes or to install other equipment which would affect the routing of the flexible shafting or the location of fastenings, the shaft may readily be accommodated to the new circumstances. The shaft can be furnished ready to install, thereby avoiding delays and special work which otherwise would be involved at the time of installation.

A typical installation on a recently built freighter is shown in *Fig.* 1. A 3½-inch bilge suction valve located just above the drain well is to be capable of operation from the shelter deck approximately 37 feet above. As the drawing shows, the handwheel on deck is not directly above the valve, also the shafting is provided with bends to allow for distortion of the supporting structure. As a result, the total length of shafting needed for this application is 53 feet 6 inches. It will be noted that the flexibility of the shaft casing permits it to be placed in the most protected position between the ship's ribs or against a bulkhead where it does not interfere with cargo stowage.

Torque to operate the valve is transmitted through a core which comprises a central wire and a series of superimposed layers of helically wound wires, each layer being wound in the opposite direction to the preceding layer. The shafting is protected by a galvanized steel casing of interlocked construction, packed with lubricant. The casing also serves as a bearing for the core, holding it in position and maintaining proper operating radius. *Fig.* 3 shows the core and case as well as the design of the end connections. Indicators fitted to the remote control station show whether the valve is open or closed.

Diameter of the core in the case of the installation in *Fig.* 1 is 1¼ inches and the shaft is capable of safely transmitting a torque of approximately 1200 inch-pounds. The angular deflection under torsion in either direction is indicated in *Fig.* 2. Selection of the proper size of shaft depends on the torque required to operate the valve and on the efficiency of the shafting, which is a function of the total length of the shafting, and of the cumulative degrees of bend.

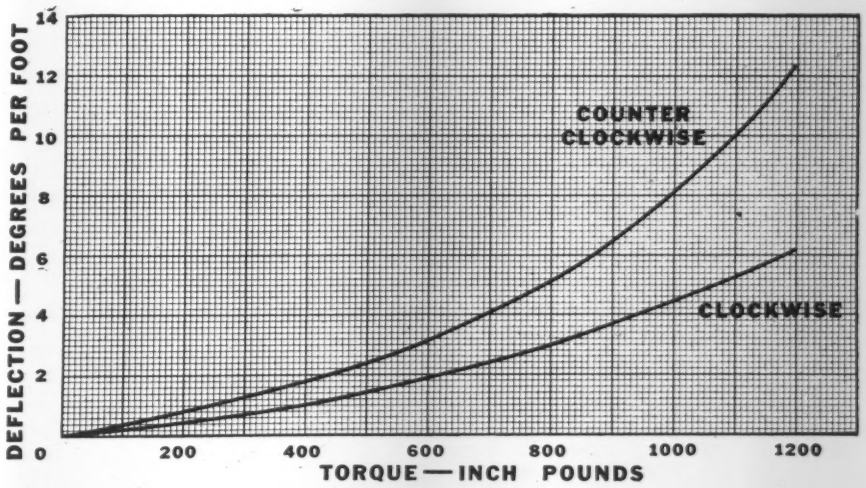
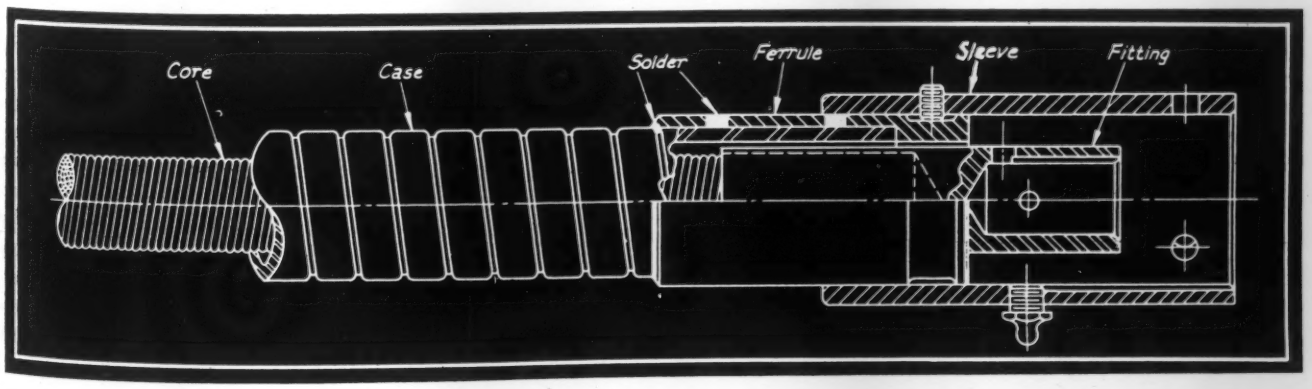
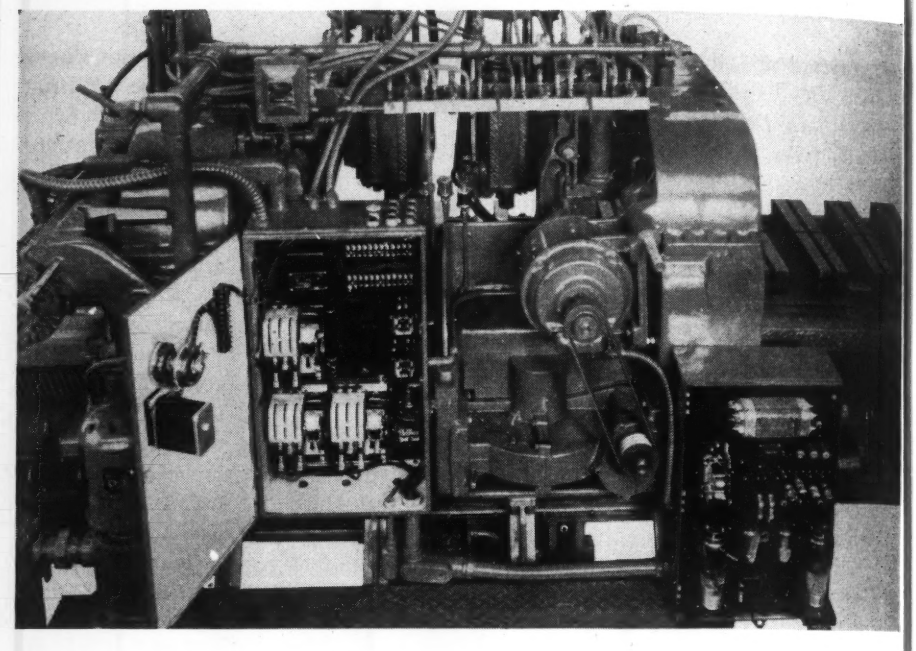


Fig. 2—Right—How the flexible shaft twists in opening and closing valves

Fig. 3—Below—Shows design of end connections for flexible shafting used in remote control of valves





Electronic Control Speeds Production

By H. G. Grondahl
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Onsrud Machine Works Inc.

WHEN the aircraft industries tooled up to put aircraft manufacture on a mass production basis, many unique machining problems were encountered. Perhaps the most unusual of these concerned the production of spar beams. It is on these beams that the wing ribs and cap strips are fastened to make the rigid framework of the wing. Not only must the spar be machined accurately to permit perfect joining of ribs and cap strips, but it also must be contoured exactly to conform with the irregular shape of the wing itself.

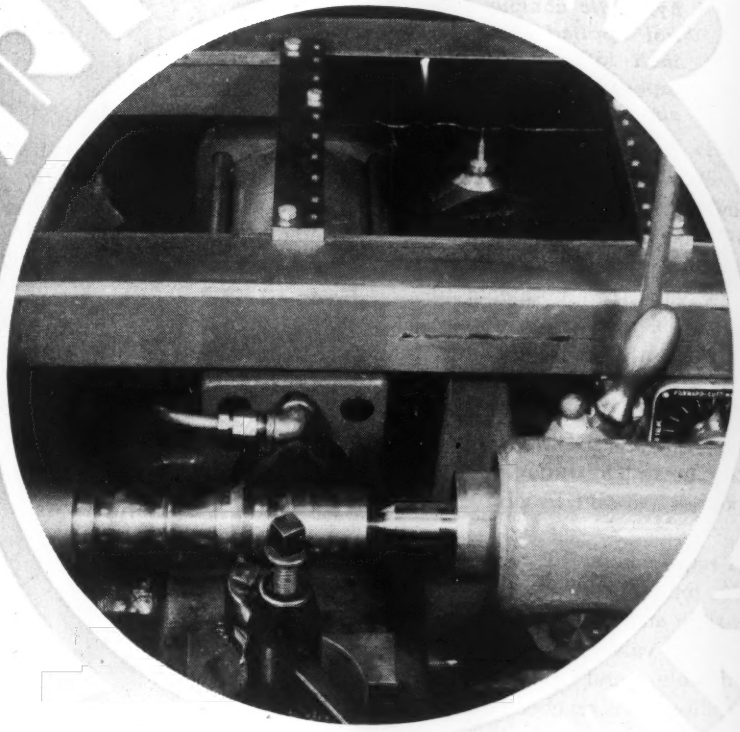
The contour milling machine selected for this type of work was designed in co-operation with Lockheed Aircraft Corp. Its advantages for this work are many. The bed and table are long enough to permit machining of the long spar in a single setup. Moving from one end of the table to the other, the carriage machines the full length of a spar when required. The carriage houses four cutter-motor assemblies—two with cutters in a horizontal plane and two in a vertical position. Each of the cutters is controlled by a follower which travels over a templet to machine accurately the contours required. One of the vertical cutters, using an ingenious pneumatic control, may be tilted off-vertical to any point up to 15 degrees to the front or back of the machine. The degree of tilt is governed by a templet and may be varied during carriage feed to machine a varying bevel or twist cut on work. Because of these four cutters, every type of cut necessitated by spar beam design may be machined—such as face, slot, and side milling, twist cutting, beveling, and making cutouts for weight reduction. Since the cutters turn at 3600 and 10,800 revolutions per minute, cutting speeds are ideal for the aluminum alloys of which spar beams are made.

Flexible carriage-speed control is required so that the cutters may be fed to the work at all times in proper relation to the ever-changing contours of a spar beam. In one pass over the table for instance, the depth of cut into the material may increase and decrease several times. The number of cutters entering the work may change from one to four. All of these varying conditions require a change of feed so that the cutter motors will not be overloaded. During a pass, points also may be reached where no cutting at all is necessary so that fast "skip" feed should be available to save time.

To solve the wide range feed demands of spar beam milling, a Thy-mo-trol drive was selected. Under this system alternating-current power is converted to direct-current to give essential stepless speed range with a rheostat-controlled direct-current carriage drive motor. This drive makes possible an infinitely variable carriage feed at any speed from 4 inches to 18 feet 6 inches per minute.

Because of the different cross sections of the spar beams, it would be impossible for an operator to judge the maximum speeds at which the carriage could be fed under the many different conditions encountered during a pass. With manual control, therefore, the natural tendency was to underfeed in order to protect the cutters and work, a procedure that resulted in unnecessary loss in production.

With automatic control, a cam bar is accurately contoured in proper relation to the work. During carriage feed, a rheostat-connected follower travels over this bar. The up-and-down travel of the follower as governed by the cam varies the control of the rheostat. Thus at every point of the pass, the mechanically predetermined carriage speed is at the exact maximum that work will permit. Spar beams that formerly required up to 13½ hours machine time are finish-machined in five minutes.



Obtaining Sensitivity with Air-Hydraulic System

By

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COMBINATION of pneumatics and hydraulics to provide extreme sensitivity is used to excellent advantage in the contour-turning unit illustrated in *Fig.* 1. Utilizing a thin metal templet as a master, the unit controls turning, boring and facing operations automatically within .0002-inch. Tool carriage feeds are hydraulic, controlled by pneumatic relays which are sensitive to air pressure in the system. This pressure varies with the movement of a tracer on the templet.

To provide for turning the tracer through 360 degrees for longitudinal turning or

for facing, it is supported by a cast-iron arm attached to the cross-slide of the lathe.

Flexing of hydraulic connections to the power cylinders is avoided by mounting both power cylinders on the carriage and by mounting the hydraulic power unit on a monorail track so that it is free to move back and forth with the carriage. By anchoring the piston rod of the longitudinal cylinder to the lathe bed, the longitudinal cylinder moves itself as well as the carriage. Cross-slide is attached to piston rod on cross-cylinder.

Fig. 1 — Pneumatic-hydraulic contour control applied to a 12-inch lathe machines tapers or complicated shapes within an accuracy of .0002-inch

The hydraulic power unit consists of an oil reservoir, motor-driven oil pump, pneumatic relay-operated hydraulic valves for both longitudinal travel and contouring action, and directional flow selector valves to provide the necessary connections for setting up the machine to turn, bore or face. Both the oil pump and the valves operate under oil.

The schematic diagram, *Fig.* 2, shows that a spring-loaded tracer increases and decreases the flow of air from the air pilot nozzle. Air from a constant-pressure source is supplied to the system through an orifice to maintain a loading pressure of 35 pounds per square inch when the tracer is in a neutral position.

This pressure is the medium by which requirements for movement of the cutting tool are transmitted from the templet and tracer to the hydraulic system, controlling both the cross-slide and the carriage movements by positioning the pneumatic relay-operated valves.

Air-loading pressure is applied to a flexible metal bellows which forces its housing against the spring until equilibrium is reached between the force of the spring and force of the air prressure on the bellows. During changes in air pressure the bellows breathes but always is in a position corresponding to the air-loading pressure. Since the hydraulic valve stem is attached to the free end of the bellows, it also moves up and down with changes in air pressure and is at any one instant in a position corresponding to the pressure.

In operation over fairly smooth contours, the air-loading pressure does not vary appreciably from the 35-pound neutral, since any tendency to depart from this pressure causes almost instantaneous movement of the cross-slide in the direction required to restore the pressure.

Longitudinal travel is

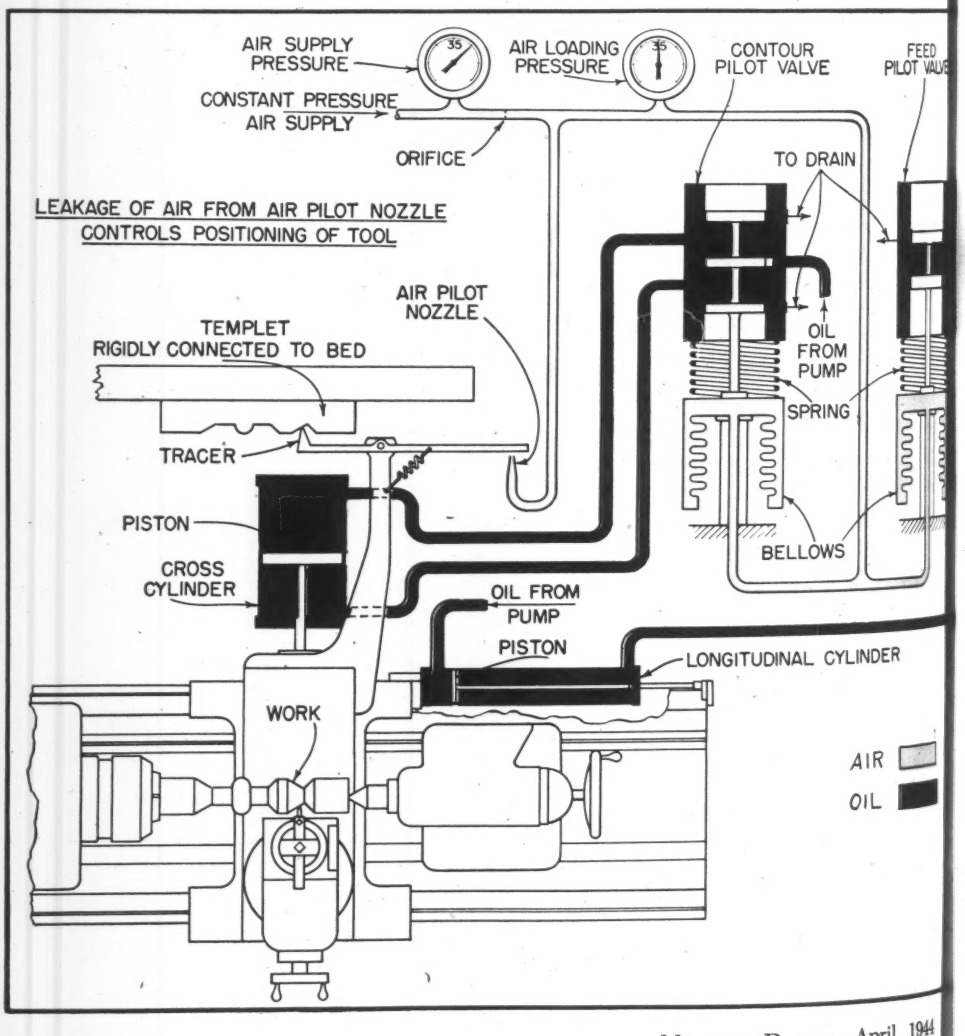
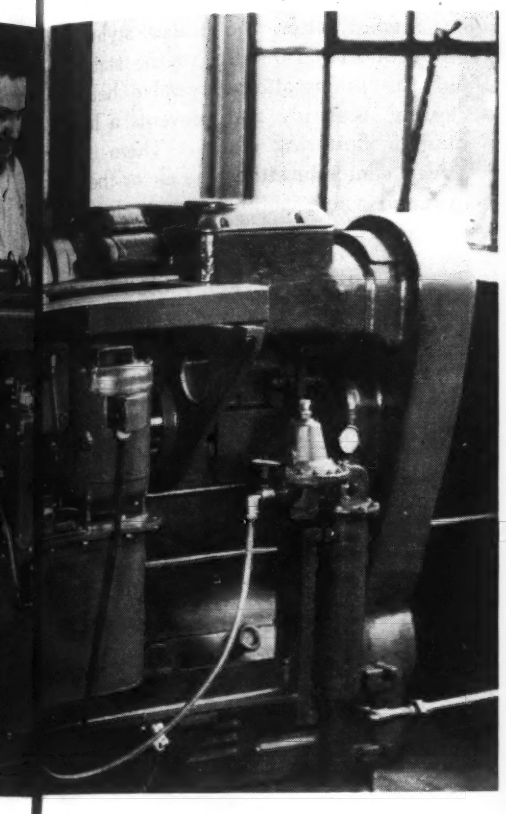


Fig. 2—Schematic diagram showing how pneumatic pilot system responds to movement of tracer and controls hydraulic valves for carriage travel and contouring action of cross-slide



controlled by the feed valve which in effect throttles the oil drain from the longitudinal power cylinder in accordance with changes in air-loading pressure. The maximum rate of drain from this cylinder and consequently the maximum rate of feed occurs when the air-loading pressure is at the 35-pound neutral. Either an increase or a decrease in air-loading pressure from this valve decreases the rate of speed. In fact if the deviation from the neutral air-loading pressure is sufficient, the longitudinal travel is stopped altogether. Such a condition exists when facing a square shoulder.

Operation of both the feed valve and the contour valve in this manner from the same air-loading pressure results in a smooth feed; for example, the feed slows down or stops when cutting a taper or a square shoulder.

To measure minute amounts of tracer stylus motion accurately and to transmit this motion rapidly to the hydraulic system it was decided to employ a pneumatic system similar to that used in precision air-measuring gages. The air control system is interposed between the stylus and the hydraulic system for the following reasons:

1. Air functions to amplify the stylus motion from 25 to 100 times at the hydraulic valves. This amplification is accomplished without backlash or friction.

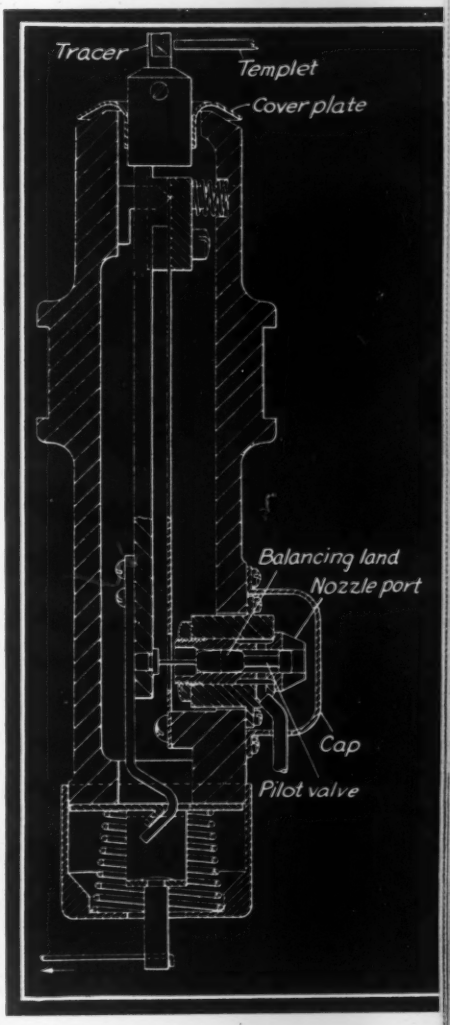
2. The air system supplies power for operation of the hydraulic valves without reaction back on the stylus. Since the air pilot valve is of fully balanced construction, a light pressure of 3 to 8 ounces is all that is required to move the stylus along the templet contours.

3. The establishment of an air-loading pressure makes it possible to control both the contouring action and the feed from the same stylus.

4. Only one low-pressure connecting line is required to the pneumatic tracer whereas several connecting lines would be required to a direct hydraulic tracer.

Detailed construction of the pneumatic tracer is shown in *Fig.* 3. A cap covers the discharge from the air nozzle so that the discharge is directed upward through the tracer body and outward between the cover plate and the body, thereby preventing the entrance of dust and dirt. The cover plate prevents the entrance of chips or other foreign material.

Fig. 3—Pivoted tracer positions the air valve to control pneumatic pressure in system



Tracer Arm Positions Pilot Valve

Shown in *Fig.* 3 in the position of no air flow is the balanced pilot valve. Under normal conditions, when the tracer arm is scanning a straight portion of profile of the templet, the tracer arm is deflected slightly toward the left and the movable balanced valve assembly is moved toward the right until the edge of the valve land has cleared the port end of the nozzle by .003 to .005-inch. In this position the land is guided by the three guides in the nozzle, and there exists an opening to atmosphere through the clearance between the valve and ports.

The balancing land of the air pilot valve, *Fig.* 3, is tapered .001-inch or more toward each end. With air under pressure in the annular chamber there is a continual constant bleed of air to the atmosphere. The taper on this land tends to center it in the bore of the sleeve, forming a lubricating film of flowing air about .001-inch thickness with the result that the balanced air pilot assembly has substantially no friction or pressure reaction regardless of whether it is mounted in a horizontal or a vertical position.

Details of the pneumatic relay for contouring action is shown in *Fig.* 4. The air-loading pressure is applied to the outside of the metal bellows and the opposing spring is connected to the bellows chamber by means of a yoke, not shown in the illustration. This type of construction, placing the spring in tension and applying pressure to the exterior surface of the bellows, is inherently stable and pro-

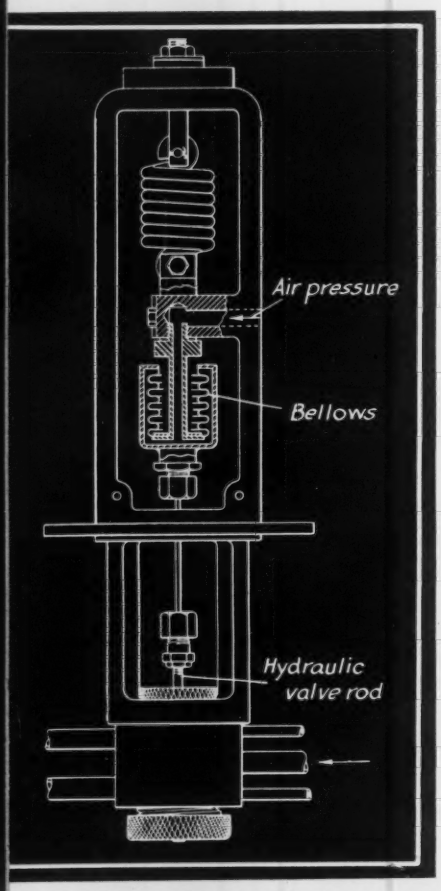


Fig. 4—Above—Pressure-sensitive pneumatic relay operates hydraulic valve for control of contouring action

vides accurate valve adjustment.

Motion of the bellows-spring assembly is transmitted to the hydraulic valve, *Fig.* 5*A*, by a small stainless steel rod with universal ball joints, and spring caps which obviate backlash. To prevent wedging of the hydraulic valve lands against one side or the other of the sleeve in which they operate, the lands are ground to have a clearance of approximately .0005-inch with the sleeve and then accurately centered in the sleeve by guides at the top and bottom. Ample lubrication of these guides is insured since the valve operates submerged in an oil bath. Drain ports are cut so that there is a small amount of drainage from both sides when in neutral position. This eliminates the effect of flow reaction as the valve settles into its final neutral position and also insures that even a slight movement from the neutral position will cause a response at the power cylinder.

This design, which deals with fluids in motion rather than starting fluids in motion after the neutral point is passed, makes possible a reaction at the power cylinder within a few thousandths of a second after the tracer stylus is moved. It also adds to the stability of the pneumatic relay-valve bellows spring assembly and prevents a hunting or fluttering action. There is no dead center in either the air or the hydraulic valve action because in each case neutral is a rate of flow.

Pneumatic Relay Used

Feed valve utilizes a pneumatic relay similar to that employed for the contour valve, *Fig.* 4. The hydraulic valve, shown in *Fig.* 5*B*, may be considered as a variable-flow resistance to the drain of oil from the feed power cylinder. It allows the maximum rate of drainage and, therefore, the maximum rate of feed when the loading pressure is at the neutral value of 35 pounds per square inch. More or less pressure restricts or closes the passage of drain.

Stem design, *Fig.* 5*B* provides for adjustment of the rate of maximum feed by turning the valve stem with reference to the sleeve. A gear mechanism controlled by knob and flexible shaft is provided for this purpose.

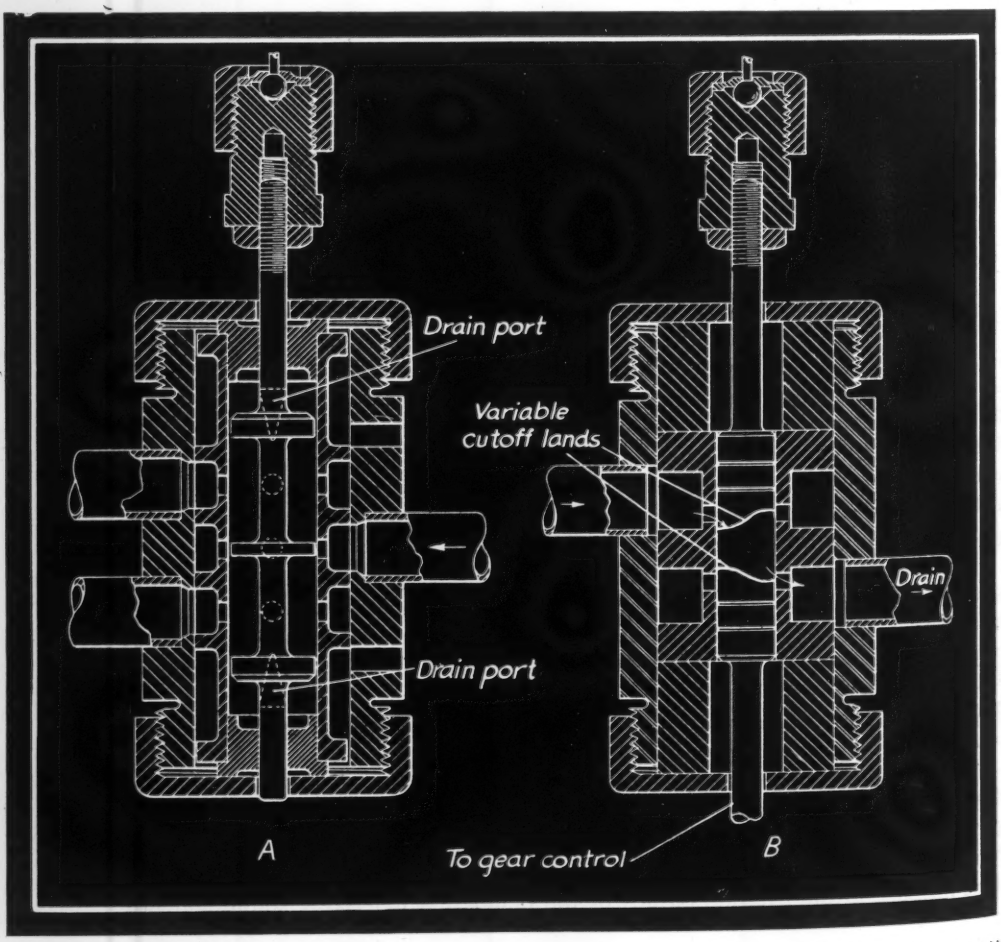
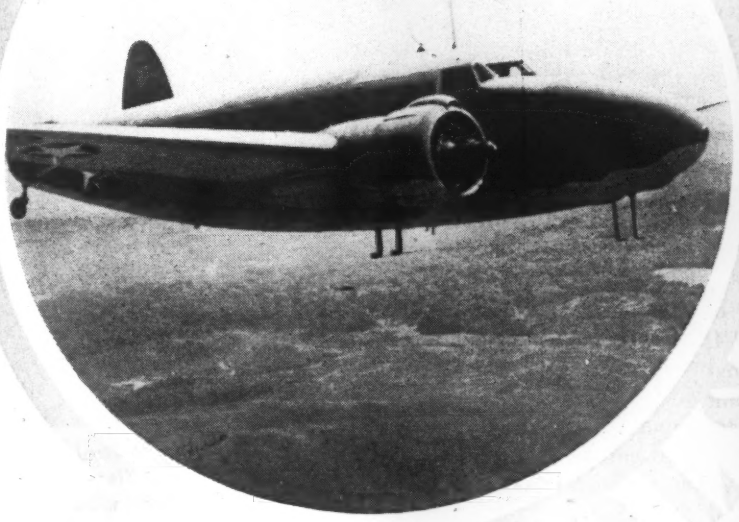


Fig. 5 — Right — Pneumatic - relay - operated hydraulic valves. At A is shown valve for contouring action, utilizing balanced design to make system sensitive to slight changes. Valve at B for longitudinal travel has variable lands on stem to preadjust rate of bleed. This is effected by rotating stem through a knob, flexible shaft, and gear arrangement



ELECTRIC Auxiliary Drives for Aircraft

By Richard M. Mock
President
Lear Avia Inc.

AT THE beginning of the war there were comparatively few electrical-mechanical installations in airplanes. However, with the increased amount of engineering time devoted to such designs, more and more installations of this nature have been made. Electrical systems from the vulnerability viewpoint offer less area susceptible to gunfire than do hydraulic systems. The rupture of a wire by gunfire usually affects only one circuit, while the rupture of a hydraulic tube may affect an entire system or a major portion of it. In addition, electrical systems can be powered by

From a paper presented at the twelfth annual meeting of The Institute of the Aeronautical Sciences.

multiplicity of wires going through the airplane by different routes and thus have any one wire able to carry normal loads and use the multiplicity in parallel to carry the overload.

Electrical-mechanical combinations make possible the design of units as complete packaged assemblies, *Figs.* 1 and 2, containing all of the switching controls, control mechanisms and relays so that for installation it is only necessary to connect the driven mechanism and the standard electrical connector. Production is facilitated as all adjustments can be made beforehand and there is no bleeding of oil lines and no need for skilled mechanics having a general knowledge of the overall assembly. The result has been simpler subassemblies, expediting production. Another rotary actuator is shown in *Fig.* 3, having a maximum output of 850 inch-pounds but weighing only 7.83 pounds.

The following is an interesting example of the development of a linear actuator, *Fig.* 4, which performs the same functions as a hydraulic retracting cylinder. The original unit of this type, built in the summer of 1942, weighed slightly over five pounds and was designed to move a peak operating load of 1000 pounds. As originally built it actually could put out 1200 pounds. With the experience gained from this first unit, a completely new design was evolved before the unit was put into production. As initially produced in quantities, the unit was designed to have a peak operating output of 1800 pounds but actually was good for 2200 pounds. Continuous study of details of the design, careful gear engineering, and improved metallurgy all coordinated with minor modifications in motor design, resulted in a unit which recently has been produced with a peak output of 3000 pounds though the weight is still the same as that of the original unit. With a slight increase in the motor, the unit has a peak output of 4000 pounds axial load. Indicative of how the current requirements have dropped per given load or how the output has increased per given current input is the fact that the original unit had a mechanical efficiency for the gearbox and screw of 18 per cent, the production unit 31 per cent, and the most recent design 41 per cent. All of the designs were with a half-depth Acme thread screw. Use of a ball bearing would increase the overall efficiency and the output still further.

Accessories such as these are used for operation of cowl flaps, wing flaps, landing gears, tabs, valves, oil-cooler shutters, engine-cooler shutters, carburetor air intake shutters, and the like. It is obvious that these are, in nearly all cases, intermittent-duty mechanisms.

In the design of such accessories an increase in efficiency is usually of value only when it results in a saving of weight, except in the case of large units where current drain is of importance. An increase in efficiency usually means an increase in output, and unless the unit is highly loaded, offering an increased safety margin, there is often

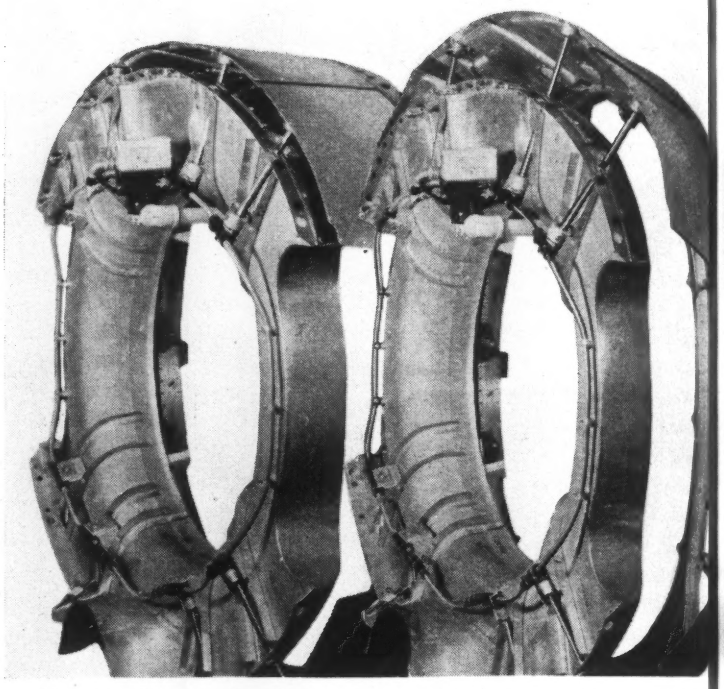
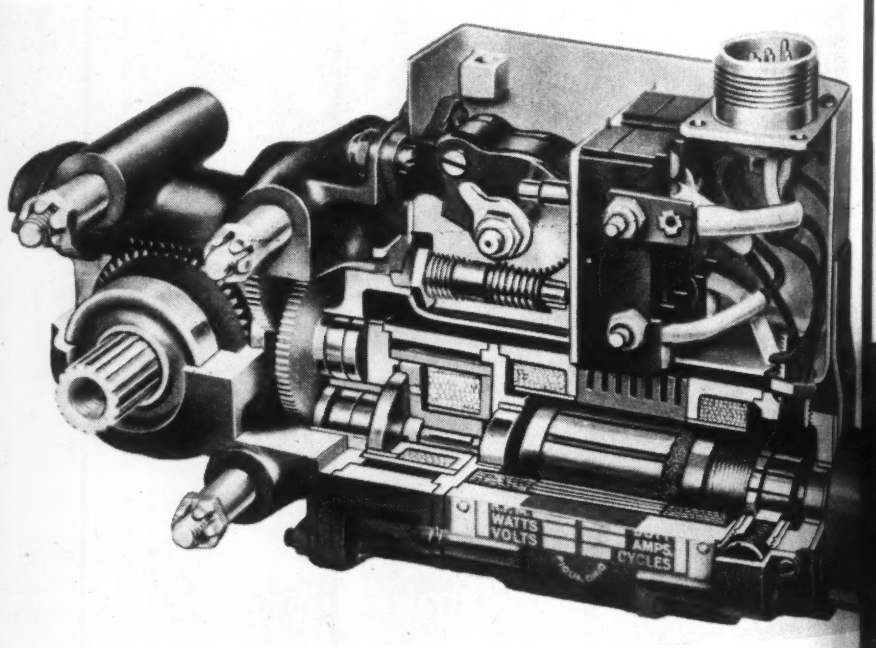


Fig. 1—Above—Application of motor control for cowl flap mechanism. Flaps are driven through flexible shafts, total weight of drive being 21.8 pounds

Fig. 2—Below—Cutaway view of rotary actuator incorporating reducing gearing and control elements such as limit switches



no practical value to the increase in efficiency. However, by continual effort to raise the efficiency, it is usually possible to substitute a smaller, similarly designed unit or to use the originally improved unit in the place of the next larger unit for another application.

As an example, on one small pursuit airplane having two linear actuators in the engine temperature control system, one for oil and one for glycol, each weighing 7.4 pounds complete with automatic temperature control, it has been possible to substitute the next smaller unit weighing only 4.79 pounds complete with automatic temperature control. Thus, in this one airplane the total saving of 5.22 pounds in the empty weight has been achieved.

Electrical-mechanical systems have always been confronted with the problem of handling inertia loads. Originally it was almost impossible to position an electrical-mechanical device accurately because of the variation in coasting and override with varying external flight loads and flight conditions such as temperature, etc. Through the use of an electromagnetic clutch to separate the high inertia parts from the lower inertia parts of the device, accurate positioning has been possible. It is our practice to install a simple magnetic clutch between the motor and the first gear train. Electrically, it is in the motor circuit. Thus, the motor armature, with its high speed and consequent high inertia, can be separated automatically from the driven mechanism whenever the motor circuit is interrupted.

By having the clutch located at the motor end of the gear train the clutch has been a minimum size, since at this point the transmitted torque is low. The gear mechanism, with its consequent lower speed and lower inertia, is stopped by a brake. Conventionally, the brake is applied by a spring which is overcome by the magnetic force when the motor is energized. In this arrangement, whenever the motor is not energized the brake is applied, which increases the manual load in devices provided for manual emergency operation. To eliminate the difficulty a second type has been developed which has the clutch energized magnetically in the conventional manner to transmit the load under power, but when the circuit is opened there is an arrangement whereby the spring is replaced by the centrifugal force of the rotating parts and the brake is applied as before, stopping the unit, after which, of course, the centrifugal force disappears and there is no longer any brake drag for manual operation.

As electrical-mechanical systems are essentially mechanical systems with an electrical power source, they offer many possibilities for mechanical emergency operation in case of electrical failure. Various basic arrangements have been designed. The simplest and most direct is to operate the unit manually by hand crank or cable. One unit which would require a manual emergency operation in only one direction has a spring-loaded drum around which there is a cable leading to a handle in the cockpit. The handle is pulled by the pilot, turning the drum, operating the mechanism. A pawl holds the driven mechanism at the end of the stroke while the spring returns the handle to the original position. Thus the handle can be used repeatedly by the pilot to move the device to the desired position. This unit has proved itself in service on aircraft and has been in production since before the war.

Control of the electrical-mechanical devices varies with the requirements. The most simple is by limit switches, which are normally incorporated in part of the gear

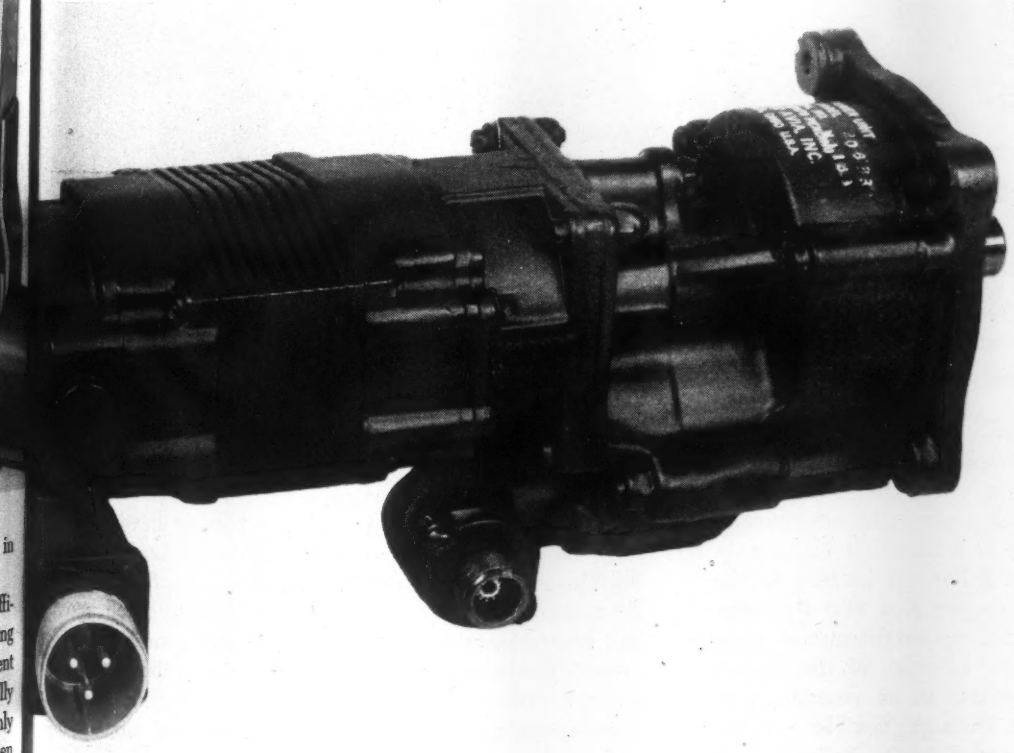


Fig. 3—Rotary actuator with built-in controls has a maximum output of 850 inch-pounds and weighs 7.83 pounds]

mechanism. It has been found best to locate these as part of the motor and gear mechanism assembly and thus eliminate the possibility of errors due to improper coordination between the switch adjustment and the driven mechanism by having these switches adjusted in a jig at the accessory factory or on the bench before installation. With this system the unit may be stopped at any intermediate position. Sometimes an electrical position indicator is used.

In some installations it is not possible to predetermine the required travel of the driven device precisely so that it is not possible to use limit switches. To achieve the desired result it is practical to have the unit run against a stop or shock absorber and then have the power shut off due to some overload device. These are of two types.

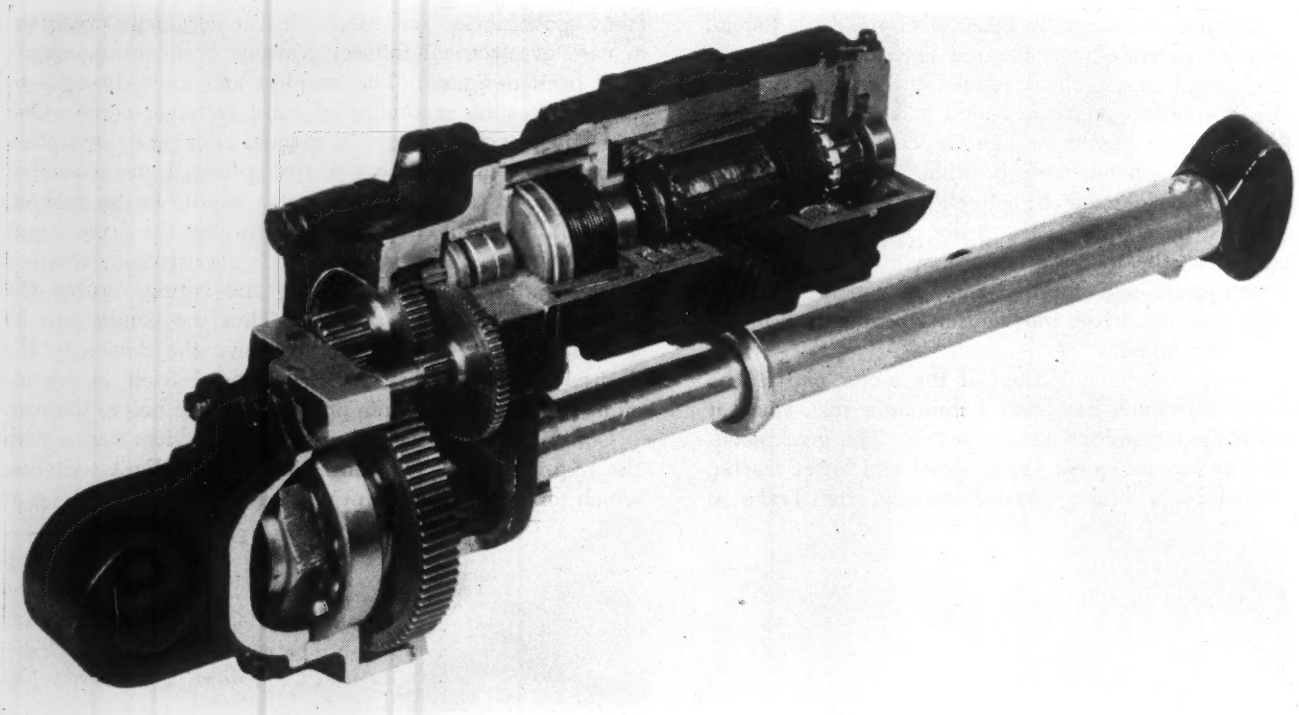


Fig. 4—Linear actuator of the type utilized for wing flaps and coolant shutters

On one the current overload due to the higher current to the motor actuates a relay. The other is the mechanical overload type where a mechanical torque limiting device moves, or a load-actuated spring device moves, actuating a cut-off switch.

Another type of control which is practical for certain applications is by a remote preset device having a mechanical follow-up. In such a system the control in the cockpit operates switches which control the actuation of the device. When the device is actuated it also moves a flexible shaft or mechanical follow-up leading to the cockpit, moving a position indicator and also the commutator or electrical contact that opens the circuit when the desired position is reached. In some of the devices of this nature it has been possible in an emergency to operate a driven mechanism through flexible shafting from the position-indicating control in the cockpit. This has been practical for tab controls and similar units.

For completely electrical remote positioning a simple bridge circuit may be used having a potentiometer on the driven unit and a polarized relay which energizes the driven device whenever the controller is moved and keeps the driven device going until it is in a position corresponding to that of the transmitter, at which time the circuit is balanced. This system is subject to some variations. Our experience indicates that it cannot be relied upon for accuracies closer than 1 per cent. Where the driven device must be positioned in smaller increments than 1 per cent of its total travel, it is felt that rather than make the relays more sensitive, the better solution is to replace the exceptionally sensitive relay by an electronic tube. By this means a system has been developed which has an accuracy of repeatability of one part in 200 with positioning increments of one part in 5000. In this system a selsyn is used both in the cockpit and on the power unit with an electronic relay located anywhere in the airplane. As electronic units are lighter than any of the wired switching or relay devices otherwise considered, they seem to be the most promising for exact positioning.

In the design of electrical-mechanical devices the use of flexible shafting has greatly extended the possibilities of such devices and simplified their installation. One might consider a gear mechanism having flexible couplings between portions of the mechanism and have the flexible coupling reach the proportions of a flexible shaft. By turning the flexible shaft at a fairly high speed, power can be transmitted at low torque in the condition analogous to the transmission of power at high voltage and low current. Power losses in the flexible shafting are approximately the same as those in a simple, single-stage spur gear train.

MACHINE Editorial DESIGN

Need Is for Increased Speeds, More Automatic Control

AS ARMAMENT production approaches nearer and nearer its ultimate peak, selective service requirements become correspondingly more stringent. So much so, that many questions now are being raised as to the ability of the nation to continue increasing its rate of production, if this should prove necessary, or even to maintain the current level.

The fact that production per manhour of certain types of military equipment has increased many times might lead to the conclusion that armament producers could readily cut down or replace their staffs without feeling the effect. Taking as an example the aircraft industry, in which production of a typical model of a fighter plane required 26,500 manhours for the hundredth plane to be built, as against 7800 manhours for the thousandth, it would seem at first glance that many men could be spared to help fill draft requirements. Actually, however, even more manpower than is available at present is needed by aircraft companies to meet the tremendously increased schedules.

The same condition—the need for more men rather than less—applies to numerous other phases of armament production. However, inasmuch as it is highly unlikely that additional men will become available, the burden of maintaining or increasing production inevitably will fall on the engineers directly and indirectly responsible for production techniques.

In this, design can continue to play an increasingly important role. Not only will it be essential that design of actual armament equipment be simplified within permissible limits, but the development of new machines for producing it will be of vital importance. Such new machines will need to be capable of even higher speeds of production and at the same time embody more of the elements of automatic control. These are factors that take on greatly added weight in view of the fact that it may be necessary, eventually, to utilize unskilled labor—both men and women—to run the various types of machines that previously only highly skilled operators could handle.

Such a challenge to design will not be disregarded. Outstanding developments in machines will go far, as in the past, toward minimizing the country's manpower problem.

L. E. Jermy

Outstanding Designs — Hig

Vacuum compartment, mounted inside the refrigerator cabinet, is sealed by a floating plate glass in the Amcoil high-altitude test chamber. Large evaporator, mounted in the upper part of the main cabinet, is isolated from the lower usable portion by a drain pan which acts as a unit of the air-duct system. On left-hand side, connected to the drain pan, is a duct into which the fan-scrolls and fans are mounted. Main refrigerator cooling is accomplished by drawing air through the evaporator and forcing it out into the usable compartment.

Capable of withstanding atmospheric pressure, the vacuum chamber is of welded construction and reinforced with I-beams so that the maximum strain at its widest span is less than five-thousandths of an inch. The front of this chamber is arranged with milled slots to receive formed-rubber gasketing. Cooling of the chamber interior can be accomplished either through the walls or in conjunction with a small evaporator which is properly baffled for forced air movement by a fan on the interior.

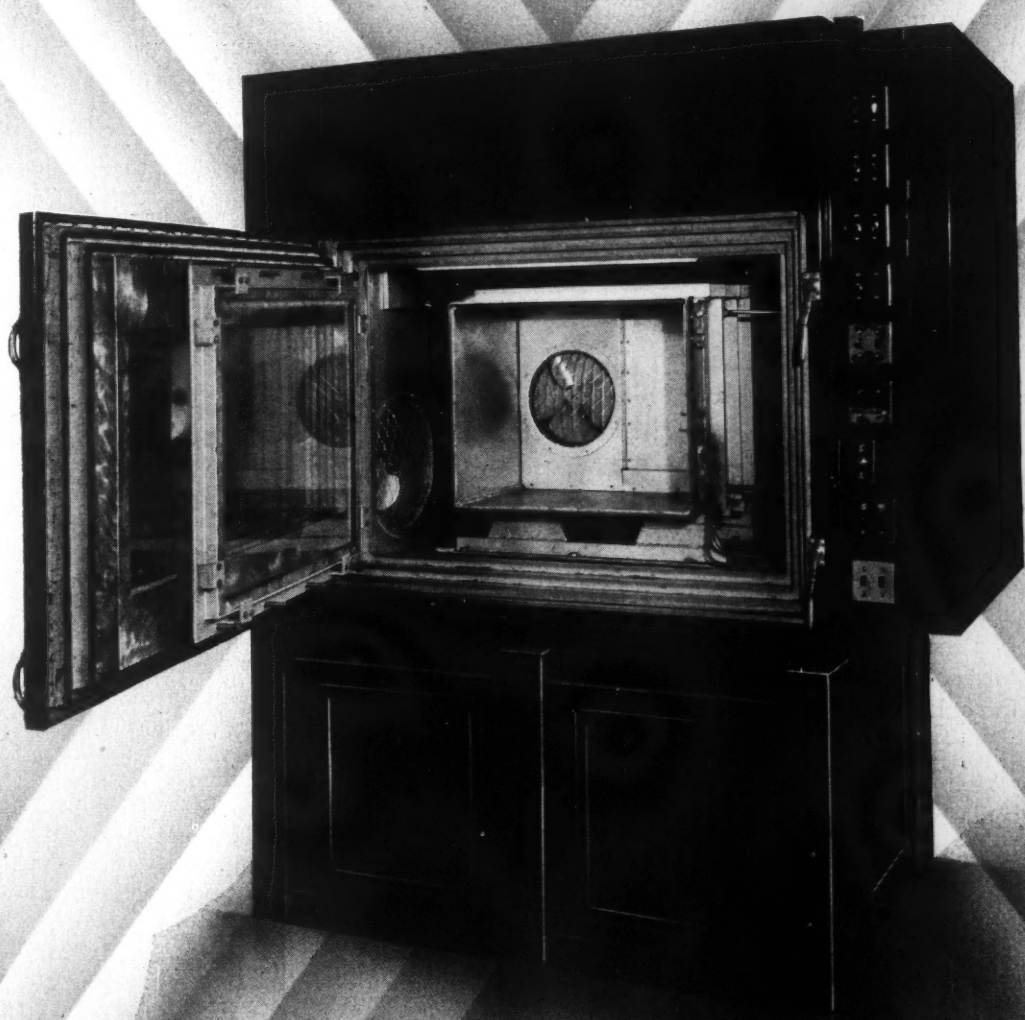
Cooling for the evaporators is accomplished with a three-stage condensing unit and the temperature range is from minus 65 to plus 70 degrees Centigrade. Heating side of the cycle is effected by strip heaters mounted both in the main chamber and on the inner vacuum chamber.

Since the main refrigerator door is not constructed to withstand atmospheric pressure, the inner chamber seal must be effected without disturbing the main refrigerator. A 1¼-inch tempered plate glass, mounted in a frame and suspended from the main refrigerator door by bolts, is allowed limited free movement on the bolts in any direction. This design assures accurate self-alignment of the glass upon sealing at the time the refrigerator door is latched and permits the glass to give freely under vacuum pressure.

Standard sylphon-type seals are used on the motor-driven shafts, being lubricated by grease, or by graphite disks on the seal faces. Comparatively close temperature tolerance can be maintained since the inner chamber is not affected by heat leakage or by fluctuation of the thermostatic controls. For general applications the thermal-bulb type of control is used.

Constructed with six inches of fiberglas insulation, the all-steel cabinet rests on a wood and steel stand provided with a removable screen for ready access to the refrigeration unit. All refrigerant lines are copper and inner and outer liners of the cabinet are cold-rolled steel, welded to prevent water vapor passage into the insulation. Refrigerant shutoff valves are all of the packless type.

High-Altitude Test Chamber



Outstanding Designs—

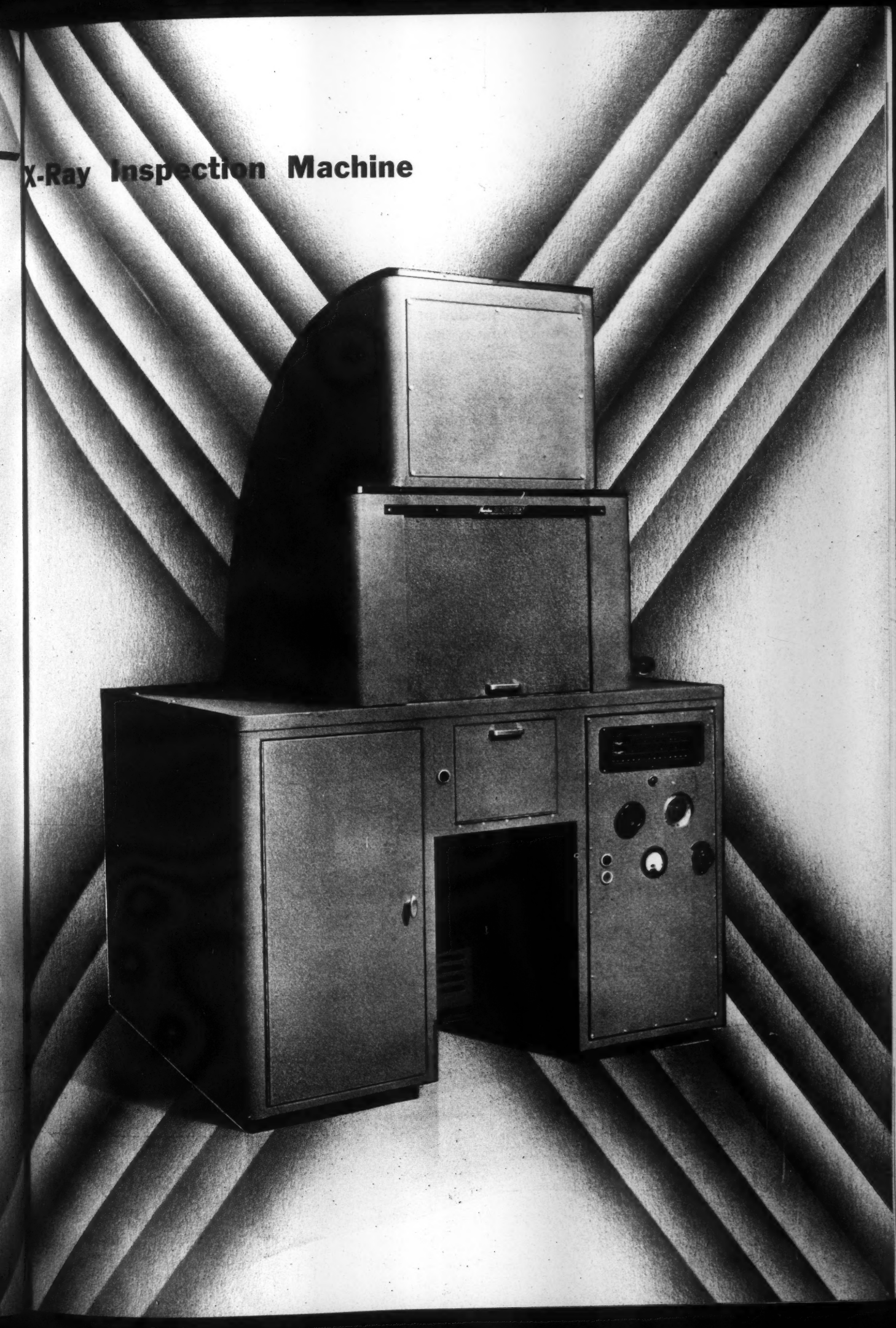
Housing of welded and braced sheet steel construction contains—in a shockproof and rayproof enclosure—all of the electrical components and controls of this X-ray inspection machine made by North American Philips. Shockproofing is accomplished by air insulation and complete grounding, while rayproofing is effected by means of lead lining in conjunction with a ray-protected X-ray tube.

Transformer for the unit is located in its base and built-in shockproof cables carry the high tension to a tube situated in a separate enclosure in the top of the unit. Radiation from the tube passes through a specimen chamber which contains the X-ray cassette or film. Radiation field is controlled by a set of manually operated shutters which limit the beam from the X-ray tube. The specimen chamber door is of counterbalanced construction, and the concealed safety switch with which it is fitted—rugged and foolproof in design—is not accessible to the operator.

Below the specimen chamber an indirect fluoroscope is arranged so that the operator may visually determine the position or other condition of the specimen in the chamber prior to radiography. The fluoroscope unit employed has an absorbing lead-glass cover over its fluoroscopic screen. Image is picked up by the operator from a mirror set at a suitable angle in the base of the fluoroscopic compartment. This design provides absolute protection to the operator from X-radiation. A unique design feature in the fluoroscopic arrangement permits the operator to view images of the specimen conveniently through a hinged viewing member equipped with a telescopic visor which, after use, disappears into the cabinet proper. For balanced design, the controls are situated in the lower right-hand section of the cabinet. Lower left-hand side of the cabinet is devoted to a storage compartment for films and accessories.

Components carried by the control panel are: Filament compensator which controls milliamperage as indicated by a meter on the panel; built-in timer with a range of from five seconds to twenty minutes—used to time radiographic exposures; and a kilovoltage control which determines the penetration of radiation and is calibrated directly in kilovolts as well as in terms of thicknesses of brass, steel and aluminum. The entire mechanism is relay-operated by an on-and-off type pushbutton.

Due to the high capacity of the X-ray tube, it is necessary to cool the tube through its oil shield. This cooling, accomplished by means of a water circulating system, is controlled by a magnetic valve which is energized from the main control setup. It effectively prolongs the life of the tube, making possible continuous operation of the unit and permitting thorough radiographic examination of the specimens.


X-Ray Inspection Machine

Outstanding Designs—Ce

Grinding wheel, regulating wheel and work rest units comprise the main elements around which design of the Landis centerless grinder is developed. While each unit is separate and complete within itself, all are correlated in such manner as to achieve uniformity in performance and appearance.

Carrying its own driving motor, spindle lubrication, wheel feed, wheel dresser, and coolant outlets for both grinding and dressing, the grinding wheel with its spindle and bearings forms a unit which is mounted on "V" and flat ways. These are lubricated from a central lubricating pump built into the machine base. Also carrying its own motor is the regulating unit, the spindle drive of which is effected by means of V-belts and silent chain. Conveniently adjustable by a rheostat mounted on the front panel of the machine—giving an infinite number of wheel speeds—the regulating wheel motor is powered by a motor generator which is mounted in the machine base.

Mounted so that it can be skewed for through feeding, the regulating-wheel spindle and drive unit can also be moved by means of a feed screw to bring it into correct relationship with the work rest which is mounted rigidly on the base of the machine. The tilt slide is mounted on a cross slide which is moved by hydraulic power parallel to the wheel spindle for dressing the regulating wheel.

The cross slide is mounted on a swivel pad with an adjusting screw built into the machine base. Grinding position of the regulating wheel is adjustable on the cross slide for alignment of the sides of the regulating wheel and the grinding wheel. Diamond for dressing the regulating wheel is attached to the work rest and is adjustable in elevation so that dressing is done on the line of contact between wheel and work.

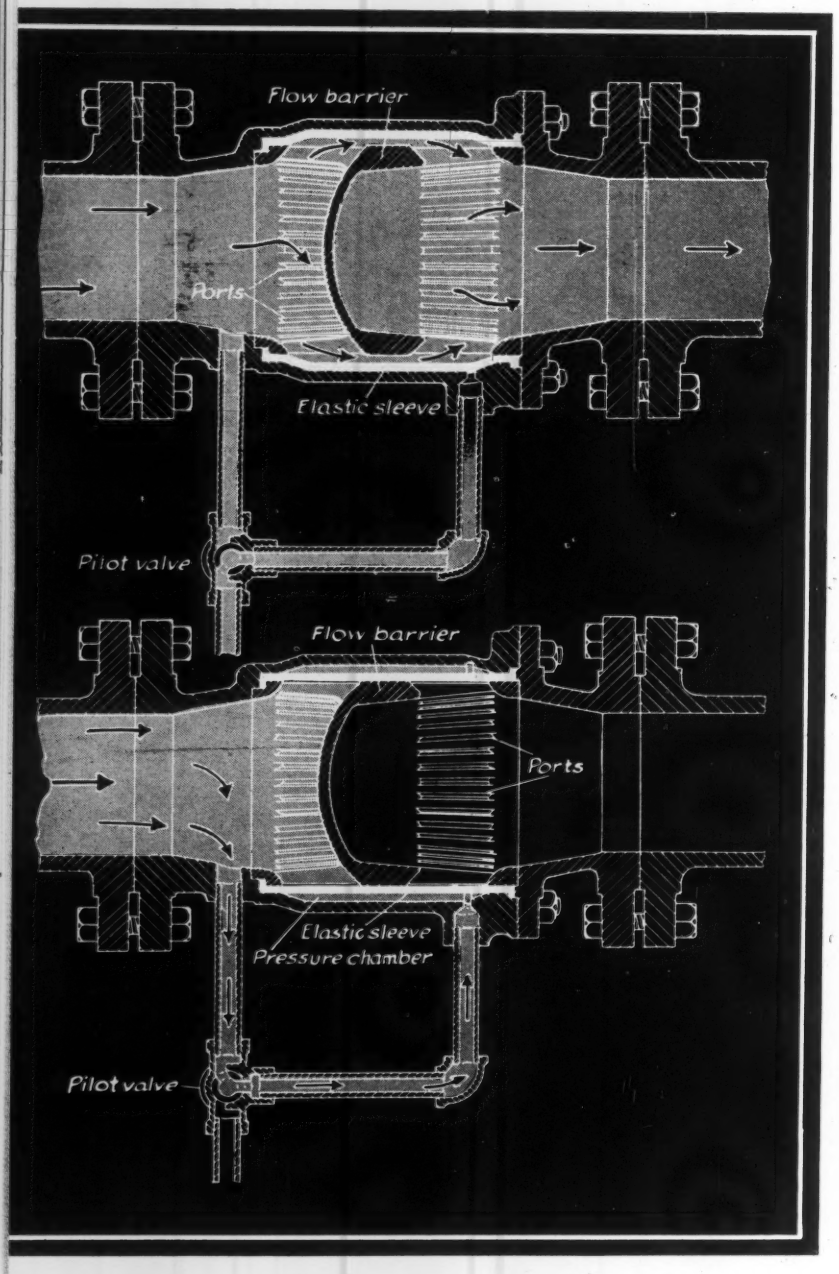
Coolant supply and motor pump unit are contained in the base of the machine. At one end of the machine is a compartment for the hydraulic reservoir, oil pump and valve units. Compartment at the opposite end, under regulating-wheel base, houses the motor-generator unit for the regulating-wheel motor. A double compartment for the electrical control panel, switches and convenient storage space for wrenches and accessories is mounted in the front wall.

Main control grouping at front of the machine bed includes a master start button and a master stop button for all electric motors as well as a three-stage motor selector switch. Both wheel dressers are controlled from the main control group. Two large selector knobs are provided and within each is a small control knob which regulates the speed of the dressing operation.

(For new machines listing see Page 208)

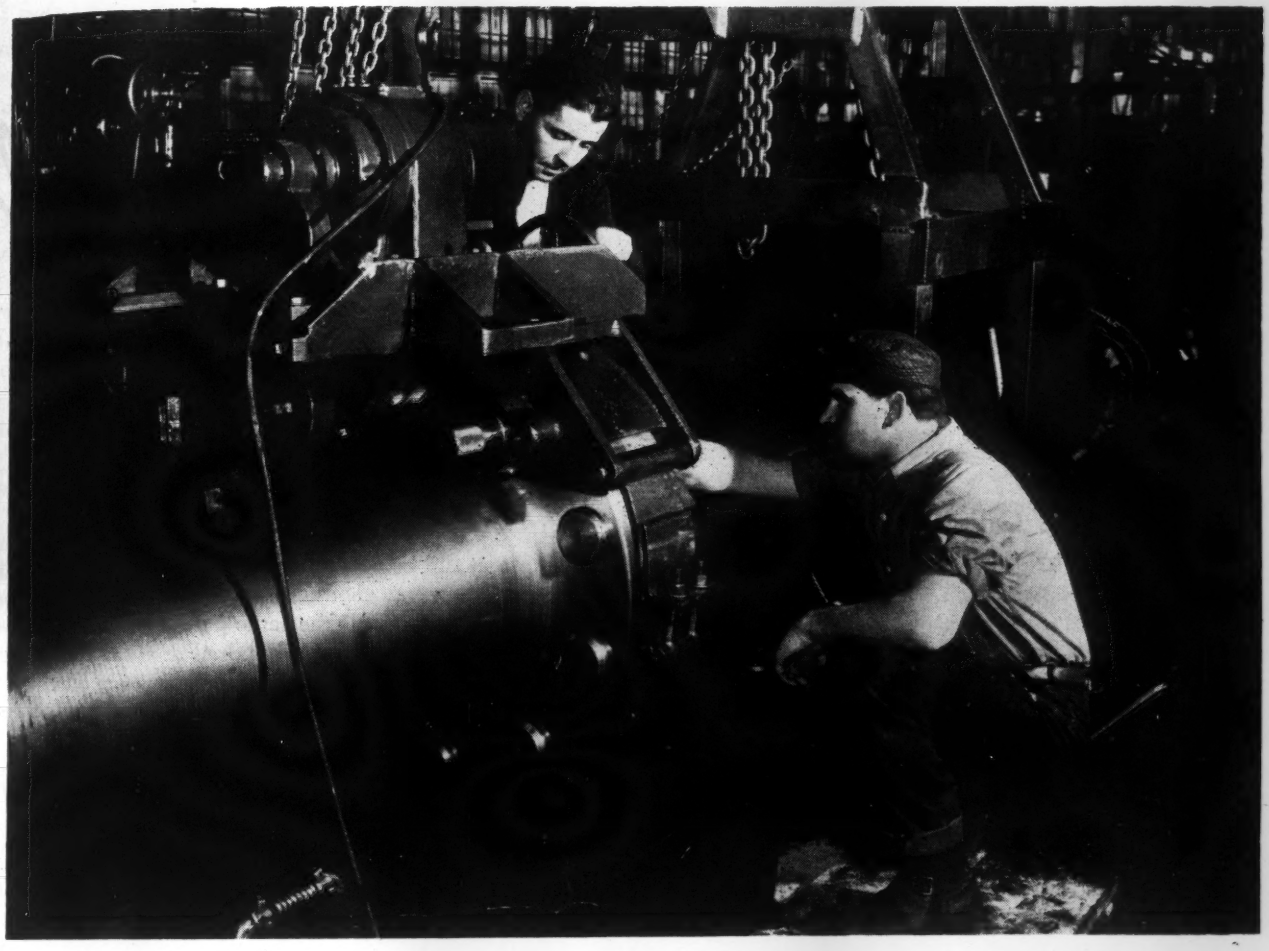
Centerless Grinder

Scanning the field for IDEA



SELF-ENERGIZING valve employs an elastic sleeve for the opening and closing of valve ports by fluid pressure in the line. Shown in the open and closed positions at left, operation of the valve is controlled by the position of its pilot valve. In the open position, the annular pressure chamber has little or no pressure. Therefore the fluid pressure expands the elastic sleeve against the valve housing and permits flow through the ports and past the barrier. In the closed position the pilot valve admits the fluid from the head side of the valve into the pressure chamber. In this way a differential pressure is built up proportional to the ratio of port area to sleeve surface, positively closing the valve. Even large valves with high working pressures may be controlled by a small pilot valve which requires little power to operate. Designed by Grove Regulator Co., the valve handles chemical solutions, oils, gases and air. A nonthrottling type, the valve is especially suitable for handling flammable materials because the only moving member is the expansible element of either rubber or synthetic.

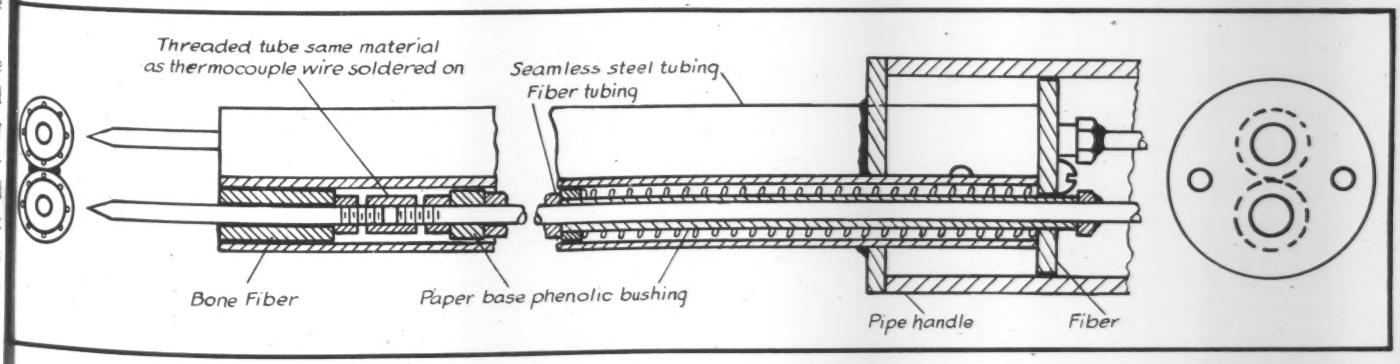
Pilot valve, often removed a distance from the main valve, may be operated manually either at the valve or remotely with a flexible shaft. Also remote operation with a solenoid valve is a simple and effective method of control. As a check valve the flow is in the opposite direction from that shown in the drawings and the pressure chamber is connected direct to the down-stream side of the valve. Then if the back pressure builds up to a sufficiently high value the valve automatically closes.



Portable machine tool for line reaming of tapered holes through shaft flanges performs a task in simple fashion that would be painstaking and time-consuming with standard machine setups. Because of its compactness and portability, reaming of the holes with shafts assembled in their machines is often possible. The reaming unit, designed by Joshua Hendy Iron Works and shown in operation above, utilizes a gear reducer drive powered by a 1½-horsepower motor mounted on the cast-iron frame of the unit. For indexing, the machine is aligned with a plug in an adjacent hole as shown. The reamers employed have spiral flutes to obviate chatter and ream to tolerances of .001-inch or less.

Improved insulation of parts provides months of service compared to days for the two-point thermocouple shown below. Developed at the Los Angeles plant of the Aluminum Co. of America the new thermocouple utilizes only two springs, one machine-made item and more effective insulation. Some of the insulating parts are stationary and some move with the thermocouple wire.

To make good electrical contact with the surface of a hot irregular object or with a surface not in line with the thermocouple, it is necessary to provide a spring for each point. Entire internal assembly is easily removable in one piece.



Choosing the Right Material

Part V—Reliability and Probability

By H. W. Gillett
War Metallurgy Committee

FINDING how to *insure* quality and freedom from scatter while the metal is a-borning is more important than how to make chemical and mechanical post-mortems after service failure has occurred. Scatter is the primary, underlying problem in the engineering use of materials. Opportunities for scatter occur all along the line.

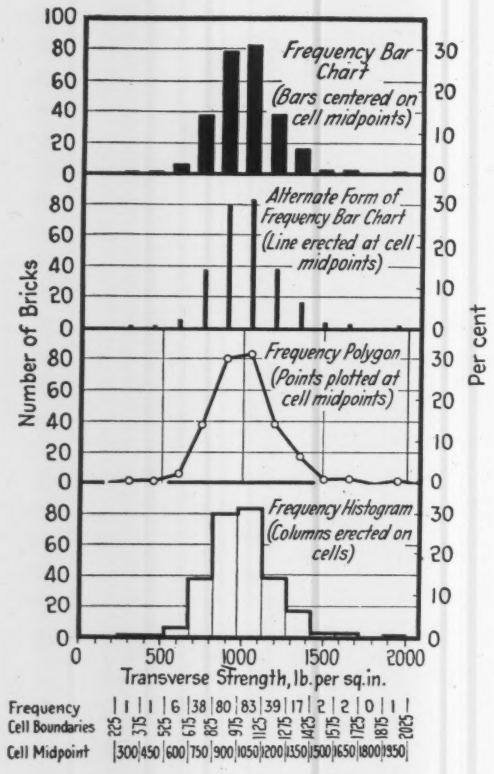


Fig. 30—Methods of plotting frequency of occurrence. The plot tends to be symmetrical. From A.S.T.M. Manual on Presentation of Data, Published in 1937

A single tensile test should be accurate within a few hundred pounds per square inch for weak, or a few thousand pounds per square inch for strong materials. But two castings off the same gate or two consecutive lengths of a bar, will probably vary several hundred or thousand pounds per square inch. If one of the castings has a blowhole or a part of the bar has a large slag inclusion or a seam, the discrepancy will be greater and, in a fatigue test, it might be very great.

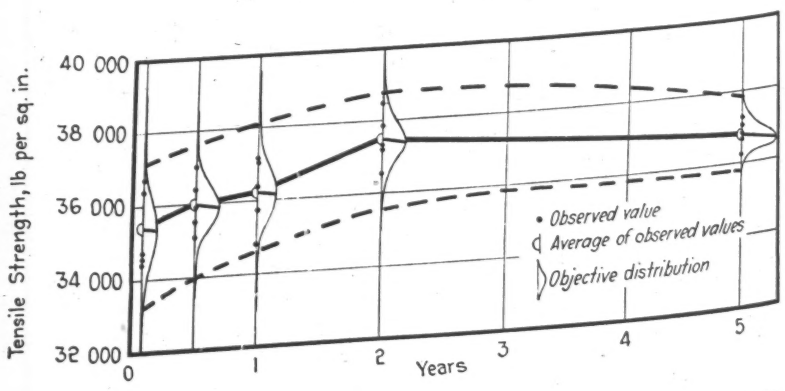
Some difference in grain size and hardenability, produced by small additions in the finishing of a heat of steel, may exist between the first and last ingot poured. Within an ingot or a casting, a little chemical segregation may occur. The top and bottom, and the outside and inside may vary a bit in composition because of the way alloys freeze.

The range of variation due to consistent and recurrent causes such as these is reasonably predictable for given plant practice. But accident may intervene. What we class as "flaws" usually come from accidents, from some unanticipated error. "Tramp" elements in the scrap used may vary widely. There can be an error in weighing some constituent of the heat. The heat may not be "finished" at exactly the same stage or tapped at the same temperature as another. Although there may be minor variations within a heat, and of course accidents may happen to any part of a heat, yet the heat is a unit, and scatter of properties from heat to heat tends to be much wider than that within a single heat.

Discussions of "aircraft quality", variation in directional properties, fatigue resistance, damage under repeated stress, notched-bar impact behavior, grain size, hardenability, and so on have brought out the fact that a heat of steel may be a law unto itself, that it responds to particular melting conditions and finishing practice that have little to do with its chemical composition.

From a War Metallurgy committee report, Part I of which appeared in the December, 1943, issue.

Fig. 31—Application of frequency principles to plotting of scatter bands. After A.S.T.M. Manual



Differences occur due to size of section. The same cast iron will be chilled in thin sections, soft in large ones. In heat-treated steel, if the section in which the test piece is quenched is smaller than that to be quenched for service and the steel does not harden clear through in the latter size, the test does not truly represent the material in service.

For any material and any processing technique there is an "average expectancy" and a range of scatter therefrom. But when the engineer makes a simulated service test to indicate the adequacy of a chosen material, he does it with an individual specimen of the material. When this specimen, say an average one, proves adequate, there still remains the question whether other specimens at the extremes of the scatter band for pertinent properties are likewise adequate. It also remains to insure that none but an adequate piece of material shall be used for the part, in spite of all accidents of production not taken into account in established normal expectancy and scatter.

The most insidious accident that can occur is a mix-up of materials in the shop. This is the nightmare of the production man, and sometimes it pays to use, say for a minor part, a material that is not otherwise justified. Devising inspection

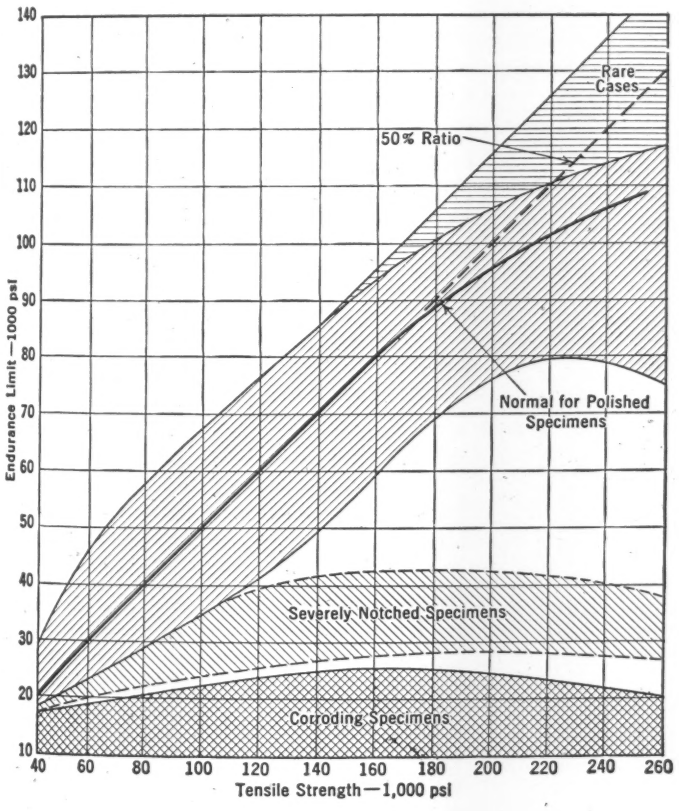


Fig. 32—Diagram indicating fatigue behavior of polished, notched or corroding specimens of steel in relation to tensile strength. This applies to ordinary corrodible steels

methods that will infallibly detect a mix-up may be of great service to the testing engineer, but mix-ups are avoidable, so the "scatter" need not take them into account.

Since 100 per cent inspection by destructive methods means zero production, it is necessary to proceed by sampling. If the sampling and testing are adequate to reveal the scatter, more samples and more tests are superfluous and wasteful. Fewer than adequate samples and tests are likewise wasteful, for they spend money in getting data that are not reliable.

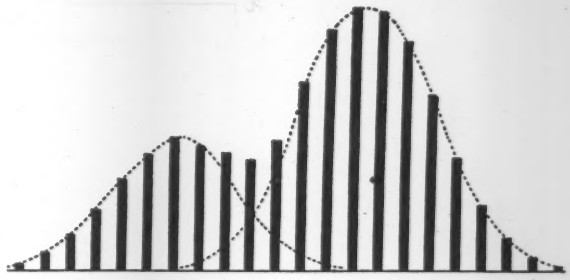


Fig. 33—Probability plot showing two humps indicating that two major causes for scatter are operating. From A.S.T.M. Manual

If two out of 1000 parts are low in the property being examined, two are high, and 996 are in the expected range, the chances that any one part picked at random will indicate that a variation in properties exists within the lot is 1 in 250, but there is no firm assurance that, if 250 are tested, one of the 4 off-quality pieces will be found.

This dilemma appears worse than it is, because even random distributions follow the "laws of chance", or probability, and engineers dealing with these problems have evolved methods of plotting and mathematical formulas that allow considerable untangling of the situation with a minimum of test data.

The first thing to do is to secure evidence about the scatter of individual observations, whether duplicate tests agree well or disagree widely, and whether successive observations, made as some variable is altered, plot in smooth curves or not.

If sufficient "duplicate" tests on a lot of material are plotted on the basis of the frequency of occurrence of a particular value, the results tend to lie in a bell-shaped, humped curve, as in *Fig.* 30. Knowing this, more can be deduced as to likelihood of scatter than is directly shown by the observations made.

Looking at the observed data of *Fig.* 31 for tensile strength of an alloy that ages, i.e., is of a precipitation-hardening type, in this light, shows that the samples taken at the start and at six months did not happen to include one with as low a value as might occur. The scatter band plotted on this basis has a regular trend, whereas one plotted to enclose the observed points would have jagged boundaries.

Often it is said that the polished endurance limit of a steel, determined on smooth specimens, is half the tensile strength. But when thousands of endurance tests are plotted against tensile strength as in *Fig.* 32, the scatter band shows that this "law" is never wholly reliable, that it becomes still less reliable above 180,000 pounds per square inch.

Looking at a scatter band, the engineer at once asks why he can't get material uniformly at the top of the band instead of having to design on the basis that some material he may get will be at the bottom of the band. Probability plotting often gives a clue to the solution of the problem. If the distribution of results is due to pure chance, i.e., if the variables affecting the results are already known and

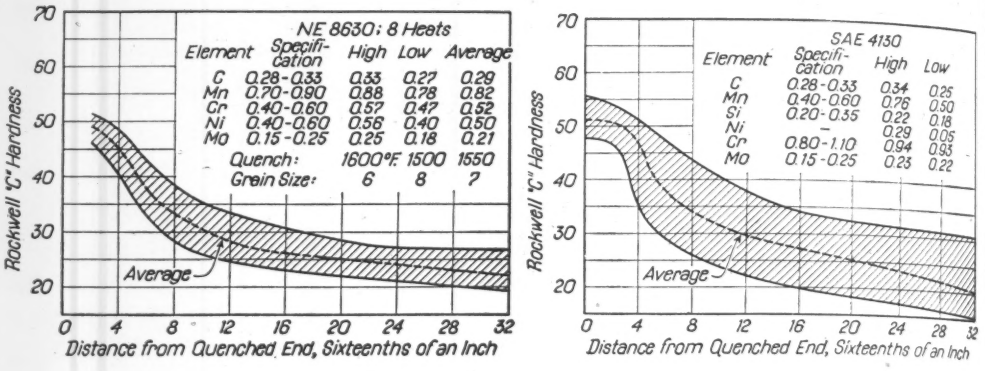

NE 8630: 8 Heats
Element Specification High Low Average
C 0.28-0.33 0.33 0.27 0.29
Mn 0.70-0.90 0.88 0.78 0.82
Cr 0.40-0.60 0.57 0.47 0.52
Ni 0.40-0.60 0.56 0.40 0.50
Mo 0.15-0.25 0.25 0.18 0.21
Quench: 1600°F. 1500 1550
Grain Size: 6 8 7
Average
Rockwell "C" Hardness
Distance from Quenched End, Sixteenths of an Inch
SAE 4130
Element Specification High Low
C 0.28-0.33 0.34 0.25
Mn 0.40-0.60 0.76 0.50
Si 0.20-0.35 0.22 0.18
Ni — 0.29 0.05
Cr 0.80-1.10 0.94 0.93
Mo 0.15-0.25 0.23 0.22
Average
Rockwell "C" Hardness
Distance from Quenched End, Sixteenths of an Inch

Fig. 34—Scatter in N.E. steel and S.A.E. steel of similar average hardenability. From "National Emergency Steels" published by A.S.M.

controlled, the probability curve will be symmetrically bell shaped, if enough observations are made.

The more sharply humped it is, the better the conditions are understood and controlled. If the scatter is wide, the hump low, but the curve still symmetrical, there are a variety of causes operating. If the curve is skewed in one direction or the other, the curve indicates that some factor is disturbing the normal distribution. If the curve has two or more humps, it definitely shows that two or more important variables are operating; *Fig.* 33.

When probability methods are applied to unpedigreed samples, taken from regular production with no lot number by which a particular piece may be traced back, a skewed or multihumped curve merely indicates that uncontrolled variables exist. If records have been kept so that the pieces showing peculiarities can be traced back to definite lots of raw materials and to definite times and conditions of processing, searching out the variables that need control is vastly facilitated.

Comparison between an unstated number of heats of 4130 and 8 heats of NE 8630 shows that the average, or expected, hardenability of each is practically identical, but that the variation between different heats happened to be much less with the NE steel. Such curves do much to overthrow the misconception that meeting a composition specification calling for an "old-line" steel means that the expected properties will be present in a given shipment. They bring out that specification on a hardenability basis would avoid such wide irregularities and would open the way for alloy-saving compositions, *Fig.* 34.

When a wide scatter of properties is found to exist, it is logical to work back through the processing, and sample and test at the various stages till it becomes apparent where the factor responsible for variation lies. Adequate testing at that point, on pedigreed specimens, should smoke out the variable that has been hiding and allow putting it under control. Or, if the cause of the variation is found to be in the raw material rather than in the processing, the supplier of the raw material then has to go through the same line of study on his own processes. Ultimately the vital locations for sampling and testing will come to light. A reasonable amount of control and testing at that location will reduce scatter, make the final product more uniform and reliable, obviate much of the later testing that would otherwise be required, and let the engineer design in confidence.

The mathematics and terminology of the science of probability are obscure to the general reader and, it must be confessed, the specialist in this field often tends to apply the methods to what his own science should tell him are inadequate data on insufficiently pedigreed samples. A general grasp of the idea of scatter and a little horse sense will lead the engineer to much the same conclusions and the same remedial actions that the specialist might arrive at by devious mathematical paths.

Dodge[1] presents a common sense, engineering discussion, without much obscure phraseology, on the utility of the probability point of view in sampling and inspection. Control charts, such as are discussed by Simon[2] and are coming into use at the U. S. Arsenals[3], can be employed to abbreviate testing. The methods used by the Arsenals are also described in publications of the American Standards Association[4], and Freeman[5] has listed some recent articles on statistical methods.

When mechanical tests are legitimately used for control in *processing*, the engineer should be on his guard for the tacit assumption that the figures are necessarily correct ones for use in *design*. The relation between properties of test bars and properties of castings is a case in point. Test bars for evaluation of cast metals are seldom taken from actual castings. They are separately poured, or at least separately gated and are standardized so that ample feeding is provided to offset shrinkage. It is stated that[6] "it is frequently possible to increase the strength of improperly gated castings, as determined by actual breakdown tests, by as much as 50 to 100 per cent through the redesign or relocation of the gates". This comment refers to aluminum castings, but is pertinent to all others. Cutting up actual castings and testing their heavy and light sections is likely to prove an eye-opener. The light sections may turn out a lot better and the heavy ones a lot worse than the "standard" test indicates.

The separately cast test bar is useful to the producer as a method of control of molten metal quality, to eliminate scatter from that cause, but it tells nothing at all about scatter after the metal leaves the ladle.

Consideration of scatter leads to realization of the necessity of making tests, and drawing specifications that are pointed at the exact properties it is desired to control, to accomplish the end in a straightforward manner, rather than by indirection.

(Continued in next issue)

[1] Dodge, H. P.—"Statistical Control on Sampling Inspection", A.S.T.M.; "Good Data", A.S.M.E., 1932; *American Machinist*, October 26 and November 9, 1932.

[2] Simon, L. E.—*Engineers Manual of Statistical Methods*, 1941.

[3] Edwards, G. D.—"Quality Control of Munitions", *Army Ordnance*, Vol. 23, 1942.

[4] *Guide for Quality Control, Control Chart Method of Analyzing Data*, 1941. *Control Chart Method of Controlling Quality During Production*, 1942. American Standards Assn.

[5] Freeman, H. A.—"Recent Publications on Statistical Methods", *Mechanical Engineering*, April, 1943.

[6] "Tentative Recommended Practice for the Casting of Aluminum Alloys in Sand Molds", American Foundrymens Assn., 1943, preprint No. 43-29.

Selecting V-Belt Drives for Maximum Economy

USE OF short center distances and large sheave diameters in V-belt drives can effect substantial savings in rubber. To aid designers in selecting the most economical drive, which is the one that loads the belts to their best horsepower-carrying capacities, the charts in this data sheet have been developed. The accompanying text explains the procedure in using these charts, and includes a numerical example to show the method

How To Use the Charts

1. Determine speed ratio (R) for the drive, using the full-load motor speed, R=Motor rpm/Driven rpm. Example: $R=1750/580\approx3$.

2. For the largest practical driven sheave diameter (D) determine the motor sheave diameter $d=D/R$. Example: $d=21/3=7$ inches.

3. From *Fig.* 1, using the nominal horsepower and the full-load motor speed, determine the number of belts required. Example: For 20 horsepower and 1750-rpm motor speed, follow the 1750-rpm line, noting the intersections with sheave sizes. Assuming the 7-inch sheave, project horizontally from the 7-inch 1750-rpm intersection to the 20-hp line, then vertically down to the number-of-belts scale. The value, to the nearest tenth, is 4.4 belts. This procedure may be repeated for a number of sheave sizes, to give a choice of drives from which the most economical may be selected.

4. Find the ideal center distance corresponding to R and D, using the multiplier given by *Fig.* 2. Ideal center distance is equal to the multiplier times the diameter of the large sheave. Example (see dotted lines on chart): For ratio $R=3$, the multiplier is 1.3, giving an ideal center distance of $1.3\times21=27.3$ inches.

5. For the center distance just found and the difference in sheave diameters ($D-d$) determine the arc of contact correction factor from *Fig.* 3. Example: For $C=27.3$ inches and

APPLICATIONS	Electric Motors										Engines		
	A.C.								D.C.				
	Squirrel Cage				Synchronous		Single Phase						
	Normal Torque Line Start	Normal Torque Compensator Start	High Torque	Wound Rotor (Slip-Ring)	Normal Torque	High Torque	Repulsion and Split Phase	Capacitor	Shunt Wound	Compound Wound	Gas and Diesel 4 or More Cyl. Above 700 r.p.m.	Steam	Line Shafts and Clutch Starting
Agitators—Paddle-Propeller	1.2	1.0	1.4	1.2	....	....	....	....	....	....	....	....	....
Brick and Clay Machinery	1.5	1.3	1.8	1.5	....	....	....	....	1.4	....	....	....	2.0
Bakery Mach	1.2	....	....	....	....	....	1.2	1.0	....	....	....	....	....
Compressors	1.4	1.4	....	1.5	1.5	....	1.2	1.2	1.2	....	1.2	....	....
Conveyors	....	1.6	1.8	....	....	....	....	....	1.6	....	....	....	1.8
Crushing Mach	....	1.4	1.6	1.4	1.4	1.6	....	....	....	1.6	1.6	....	1.6
Fans and Blowers	1.6	1.6	2.0	2.0	2.0	2.0	....	....	1.4	....	1.6	1.5	1.5
Flour-Feed and Cereal Mill Mach	1.4	1.4	1.6	1.4	1.4	....	....	....	....	....	1.8	....	....
Generators and Exciters	1.2	....	....	....	....	....	....	....	1.2	....	2.0	1.4	1.4
Laundry Mach	1.2	....	....	....	....	....	....	....	....	1.2	....	....	....
Line Shafts	1.4	1.4	....	1.4	1.4	2.0	1.4	1.4	1.4	1.4	1.6	1.6	1.6
Machine Tools	1.2	....	....	1.4	....	....	1.2	1.2	1.2	1.2	....	....	....
Mills	....	1.6	1.6	1.4	....	....	....	....	....	1.4	....	....	1.6
Oil Field Mach	1.2	1.2	1.4	....	....	....	....	....	1.4	1.4	1.4	1.4	1.6
Paper Machinery	1.5	1.4	1.8	1.5	1.6	1.8	....	....	1.5	1.5	....	....	1.8
Printing Mach	1.2	1.2	....	1.2	....	....	....	....	1.2	1 2	....	....	....
Pumps	1.4	1.4	1.4	1.6	1.6	1.8	1.2	1.2	1.2	....	2.0	2.0	....
Rubber Plant Machinery	1.4	1.4	1.4	1.4	....	1.8	....	....	....	....	....	2.0	....
Screens	1.2	1.2	1.4	....	....	....	....	....	....	....	....	....	....
Textile Mach	1.6	....	1.8	....	....	....	....	....	....	....	....	....	....

Table shows typical service factors for various conditions

The charts and information from which this data sheet has been prepared were furnished through the courtesy of the Allis-Chalmers Mfg. Co.

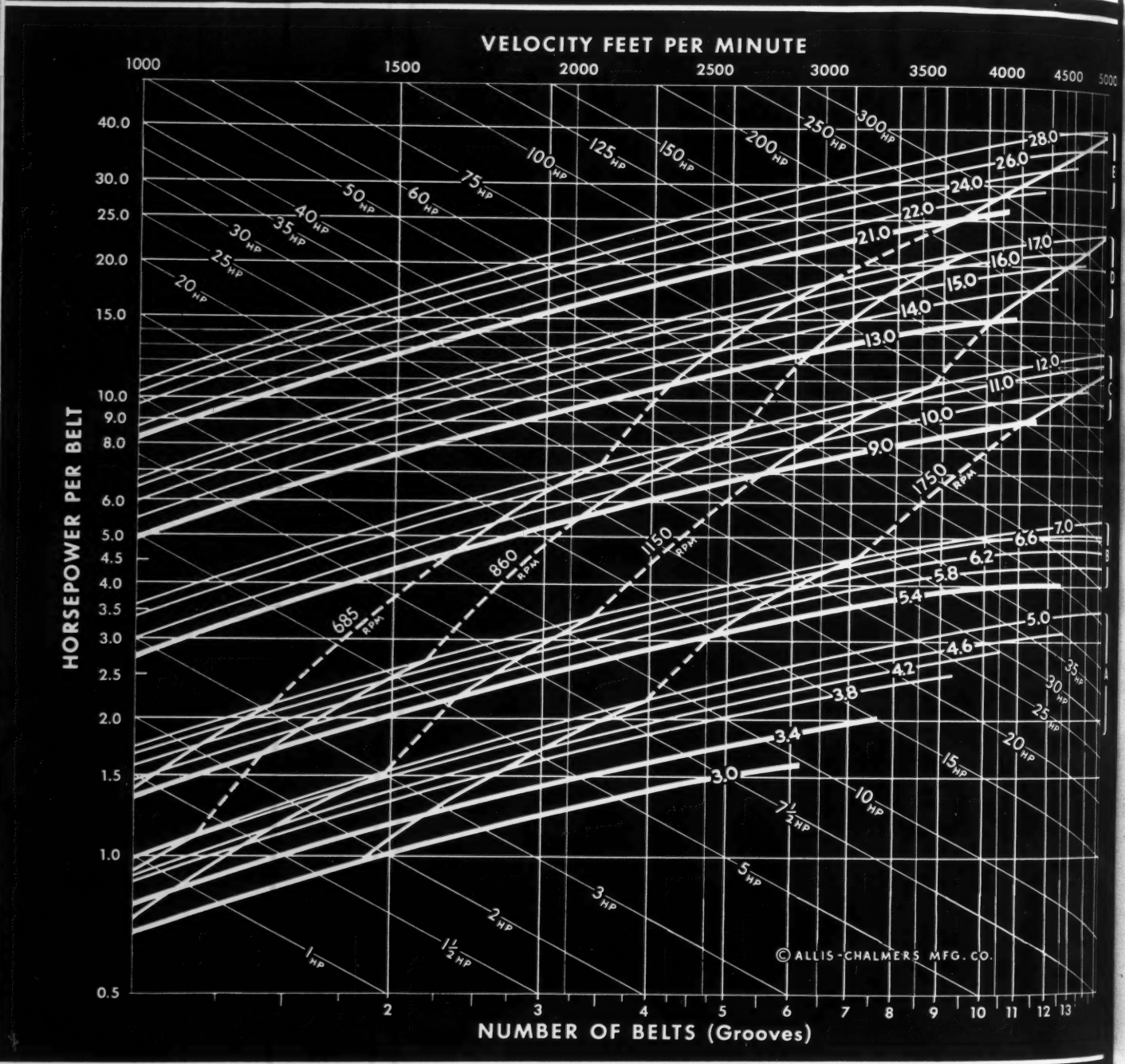


Fig. 1—Sheave selector chart (Econograph) gives horsepower per belt as a function of motor speed and sheave diameter. For example, a 7-inch pitch diameter sheave running at 1750 revolutions per minute with a size B belt will transmit 4.5 horsepower. Multiple belt calculations are facilitated by the addition of lines of constant horsepower

$D-d=21-7=14$, the factor is .92.

6. Determine total correction factor, which is equal to the service factor (see table) divided by the arc of contact factor just found. Example: Assuming a compressor driven by a wound-rotor induction motor, service factor = 1.5, the total correction factor = 1.5/.92 = 1.63.

7. Multiply number of belts already found under paragraph 3 by the total correction factor, to give the actual number of belts required. Example: $4.4 \times 1.63 = 7.17$, or seven belts.

8. Determine the exact center distance and belt length for a standard belt, using the belt selector chart, Fig. 4. From small sheave diameter project horizontally to intersection with ratio line, then vertically to one of the standard belt length lines, then horizontally to the center distance. Example (see dotted lines on chart): From the intersection of the 7-inch sheave line and the 3-ratio line, a vertical intersects the 105-inch belt length line at a point corresponding to a center distance of 30 inches, which is close to the value 27.3 previously found as the ideal.

9. Repeat this procedure for a number of different sheave sizes, and select the most economical.

ENGINEERING DATA SHEET

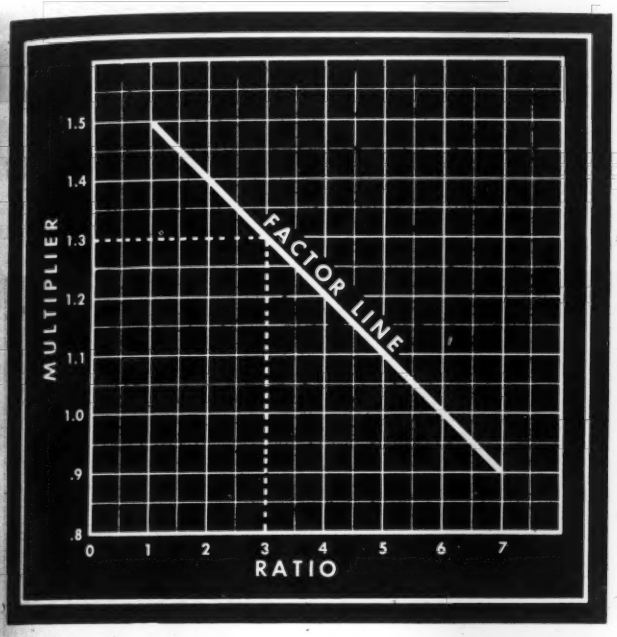


Fig. 2—Ideal center distance is equal to the multiplier times the diameter

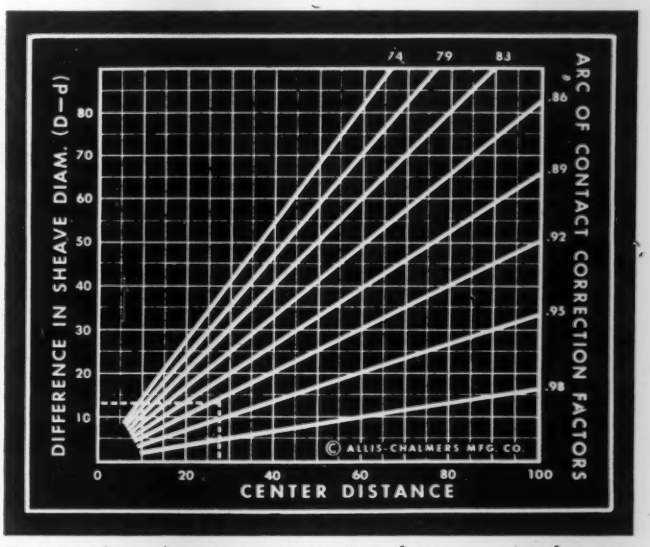


Fig. 3—Arc of contact correction factors as a function of center distance and difference in sheave diameters. Values may be read to the nearest one-hundredth by interpolation between lines

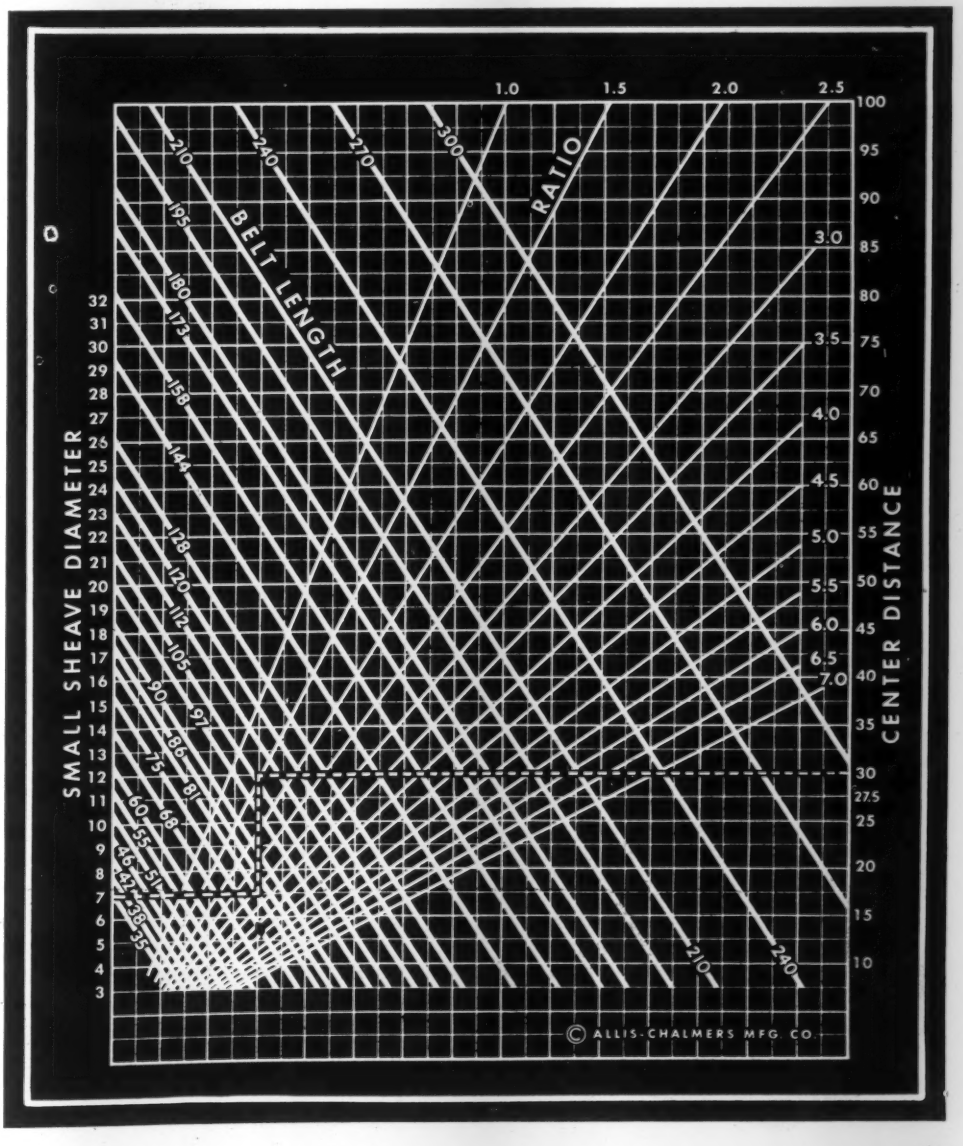


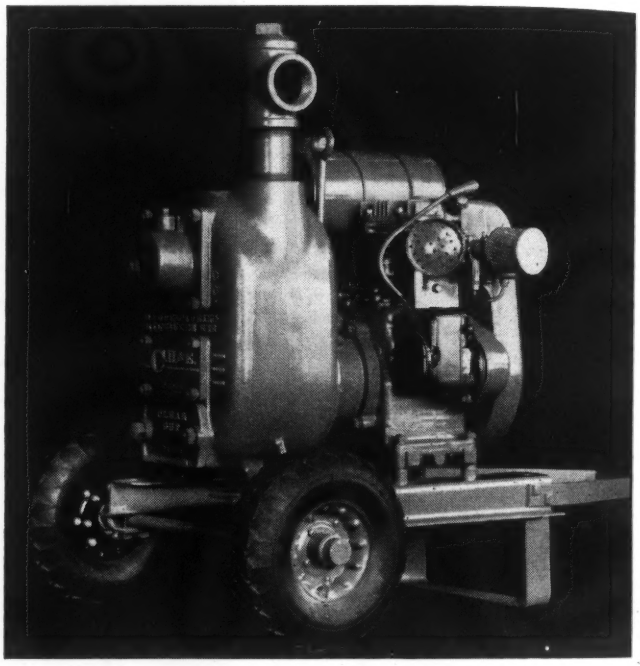
Fig. 4—Belt selector chart gives center distances for standard belt lengths and for various ratios and sheaves

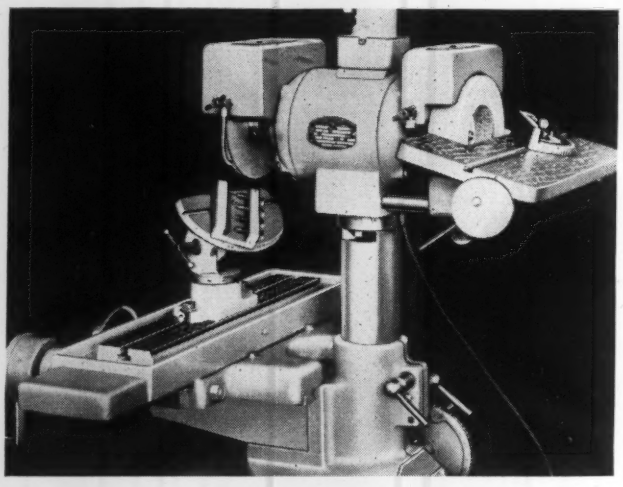
Applications

of Engineering Parts, Materials and Processes

Close-Connected Engine Drive

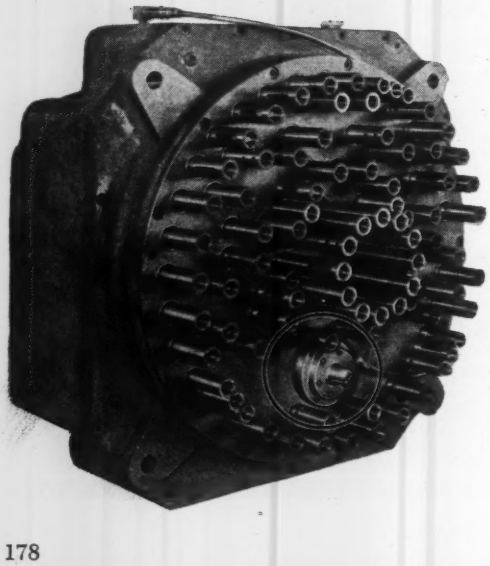
IN THE engine-driven pump assembly shown at right the crankcase of the Wisconsin engine is close-connected to the volute of the self-priming centrifugal pump, permitting the pump impeller to be mounted directly on the end of the crankshaft. In addition to compactness this design results in production and servicing economies through the elimination of couplings, belts, pump bearings, and impeller shaft. The hollow connecting casting provides a grease chamber for the pump seal, which is of the rotary type. Driving engine is of the heavy-duty air-cooled type and is equipped with roller bearings at both ends of the crankshaft; automatic choke; and a magneto with an impulse coupling for quick starting.





Motor Becomes Machine Head

INTEGRAL design of the spindle motor into the Lee universal carbide tool grinder is illustrated in the view shown at left. Mounted on the right-hand end of the motor is a six-inch recessed grinding wheel, together with a graduated tilting table and indexed protractor, for rough grinding of carbide-tipped single-point tools. The left side of the machine is for grinding chip breakers and is equipped with a universal work-holding fixture. The reversible motor is provided with a heavy shaft and dustproof bearings, the reversing switch being conveniently located on top of the motor. The column which carries the motor is capable of seven inches vertical movement and the wheel head swivels 360 degrees.



Lubricating 84-Spindle Drill Head

LUBRICANT for the gears in the Buhr 84-spindle drill head shown at left is supplied by a Pioneer Rollway pump, indicated in the circle. Mounted on a vertical hydraulic drill press, the head is used for drilling all holes in a supercharger cover. The pump is of the slow-speed positive-displacement type primarily designed to handle liquids with lubricating properties. It is capable of developing high vacuum, permitting it to be set considerably above the surface of the liquid to be pumped without danger of loss of prime.

MACHINE DESIGN'S

Materials Work Sheet

Filing Number 5.01

Stainless Steels

AISI Types 410 and 416

AVAILABLE IN: (410) Sheet, strip, plate, bar, rod, forging billets, tube rounds, tubing, welding electrodes and wire core, wire, screws, bolts, rivets, cold-drawn shapes, structural shapes.
(416) Bar, rod, forging billets, wire, screws, bolts, rivets, cold-drawn shapes.

ANALYSES:

Type	C	Mn	P	S	Si	Cr
410	.15 max.	1 max.	.04 max.	.04 max.	1 max.	11.5-13.5
416	.15 max.	1 max.	*	*	1 max.	12.0-14.0

*P or S or Se, .07 min.; Mo or Zr, .6 max.

PROPERTIES

Data applies to both types unless noted.
Values are typical unless noted.

TENSILE STRENGTH

(*psi*)

Plate, annealed†	75,000	(410)
Sheet, annealed†	70,000	(410)
Strip, cold-rolled 40 per cent†	125,000	(410)
annealed†	78,000	(410)
Wire stock		
½ to 2-in. diam., annealed	75,000	
cold-drawn	80-115,000	(410)
cold-drawn	80-100,000	(416)
Bar stock		
½ to 2-in. diam., annealed	75,000	
cold-drawn	80-115,000	
hardened and tempered	80-185,000	
Over 2-in. diam., annealed	75,000	
hardened and tempered	80-185,000	
Tubing, annealed†	72,000	(410)
Forgings, annealed†	85,000	(410)
annealed†	89,000	(416)

YIELD STRENGTH IN TENSION

(*offset* = .2%) *psi*

Wire stock		
Up to ½-in. diam., annealed	40,000	
cold-drawn	50-105,000	(410)
cold-drawn	50- 90,000	(416)
Bar stock		
½ to 2-in. diam., annealed	40,000	
cold-drawn	50-100,000	
hardened and tempered	50-135,000	
Over 2-in. diam., annealed	40,000	
hardened and tempered	50-135,000	

ULTIMATE SHEARING STRENGTH

(*psi*)

Wire stock		
Up to ⅝-in. diam., annealed	55,000	
cold-drawn	65,000	
hardened and tempered	70-105,000	

†A.S.T.M. values.

MACHINE DESIGN is pleased to acknowledge the collaboration of Rustless Iron and Steel Corporation in this presentation.

FATIGUE ENDURANCE LIMIT

(*psi*)

Bar stock	
Over ½-in. diam., annealed	38,000
cold-drawn	40- 55,000
hardened and tempered	40- 90,000

ELONGATION IN 2 INCHES

(*per cent*)

Plate, annealed†	30	(410)
Sheet, annealed†	26	(410)
Strip, cold-rolled 40 per cent†	3	(410)
annealed†	28	(410)
Wire stock		
Up to ½-in. diam., annealed	28	
cold-drawn	15-25	(410)
cold-drawn	13-23	(416)
Bar Stock		
½-in. to 2-in. diam., annealed	30	
cold-drawn	15-25	(410)
cold-drawn	11-21	(416)
hardened and tempered	12-25	
Over 2-in. diam., annealed	25	
hardened and tempered	10-20	
Tubing, annealed†	30	(410)
Forgings, annealed†	25	(410)
annealed†	23	(416)

REDUCTION OF AREA

(*per cent*)

Plate, annealed†	60	(410)
Bar stock		
½ to 2-in. diam., annealed	70	(410)
annealed	60	(416)
cold-drawn	60-70	(410)
cold-drawn	45-60	(416)
hardened and tempered	50-70	(410)
hardened and tempered	35-60	(416)
Over 2-in. diam., annealed	65	(410)
annealed	55	(416)
hardened and tempered	40-60	(410)
hardened and tempered	30-50	(416)
Forgings, annealed†	60	(410)

Materials Work Sheet

IZOD IMPACT
(ft lbs)

Bar stock (at room temperature)			
⅝ to 2-in. diam.,	annealed	100	(410)
	annealed	70	(416)
	cold-drawn	40-90	(410)
	cold-drawn	5-35	(416)
	hardened and tempered‡	30-90	(410)
	hardened and tempered‡	20-65	(416)
Forgings, annealed†		85	(410)

‡Tempering in the range of 800-1000 degrees Fahr. may result in lowering of impact strength and corrosion resistance.

IZOD IMPACT VALUES *(ft lbs)* AT SUB-ZERO TEMPERATURES
(bar stock ⅝ to 2-in. diam.)

Condition	Type	Brinell Hardness	—Degrees Fahrenheit— 90[3]	0	−50	−100
Stress-relieved[1]	410	375	54	31	24	19
Tempered[2]	410	229	91	30	28	28
Stress-relieved[1]	416	375	23	23	18	17
Tempered[2]	416	229	37	31	23	18

[1] Treated 1800 deg. Fahr. ½ hour; oil quenched plus 600 deg. Fahr., 2 hours; air cooled.
[2] Treated 1800 deg. Fahr. ½ hour; oil quenched plus 1150 deg. Fahr., 4 hours; air cooled.
[3] Room temperature at time of tests.

CREEP STRENGTH (410)
(for life of 10,000 hrs. with 1% elongation)

Temperature	psi
1000 deg. F	12,000
1100	4,500
1200	2,000
1300	1,750

HARDNESS

ROCKWELL

Sheet, annealed†	B75	(410)
Strip, cold-rolled 40 per cent†	C23	(410)
annealed†	B80	(410)
Wire stock		
Up to ½-in. diam., annealed	B80	
cold-drawn	B85-100	
Tubing, annealed†	B80	(410)

BRINELL (3000-kg load, 10-mm ball)

Plate annealed†	135	(410)
Bar Stock		
½ to 2-in. diam., annealed	150	
cold-drawn	180-240	
hardened and tempered	175-385	
Over 2-in. diam., annealed	150	
hardened and tempered	175-385	
Tubing, annealed†	150	(410)

PHYSICAL CONSTANTS[§]

Melting Point (deg F)	2700-2730
Temp. Coef. of Resistance (ohms per ohm per deg C) 0 to 100 deg C	.0015
Specific Gravity	7.8
Weight (lb per cu in)	.28
Modulus of Elasticity (psi)	29,000,000
Coef. of Thermal Expansion (per deg F)	
0 to 200 deg F	.0000061
0 to 1000 deg F	.0000072
Thermal Conductivity (g-cal/sq cm/sec/deg C/cm)	
At 212 deg F	.059
1000 deg F	.064
Electrical Conductivity (% of Int'l. Ann'ld. Copper Std.)	3
Electrical Resistivity	
Microhms per cu in at 70 deg F	22.5

§Are approximately the same for types 410 and 416.

APPLICATIONS

Type 410 is extensively used for screws, bolts, valve parts, turbine blades, pump rods, pistons and similar applications requiring a combination of high strength and corrosion resistance. It is used where dampness, alone or coupled with chemical pollution, would deteriorate ordinary steel, or where inaccessibility or wear make paint protection impractical as in grain elevator chutes, coal handling equipment and trash screens.

Type 416, as its properties would suggest, is used for screws, valve parts, pump shafts, and similar applications where processing involves considerable machining. It also is of outstanding value for applications where service involves friction under high pressures, or at elevated temperatures.

CHARACTERISTICS

Type 410 is a straight chromium hardenable type of stainless steel. It possesses good corrosion-resisting properties in all conditions of heat treatment, and has fair resistance to elevated and subzero temperatures. A wide range of physical properties may be obtained by heat treatment. It is readily forged, cold worked and machined. It is strongly magnetic.

Type 416, with minor exceptions, is comparable in physical, tensile and corrosion-resisting properties to Type 410. This alloy also is strongly magnetic. By addition of sulphur or selenium, free-cutting properties are imparted to it. The sulphur-bearing composition is most widely used. However, in special instances where both forging and machining requirements are equally important, the selenium-bearing alloy has found favor.

FABRICATION

MACHINABILITY:

Type 410 machines best with an intermediate hardness and when in the cold-drawn condition. When machined, a characteristic tough and stringy chip is produced.

Type 416 can be machined with standard shop tools at speeds approaching those used for bessemer screw stock. It cuts freely, frictional resistance is low and chips are brittle.

PROPERTIES AT ELEVATED TEMPERATURES
(short-time tests on type 410)

	Temperature (degrees Fahr.) 400	600	800	1000	1200	1400
Tensile Strength (psi)	77,500	72,500	67,500	50,000	25,000	10,000
Elongation in 2 inches (per cent)	27	25	25	30	48	68
Reduction of Area (per cent)	73	70	70	77	92	95

Maximum machinability is attained when the stock is tempered to an intermediate hardness and moderately cold drawn.

With both types 410 and 416, sulphur-base oils are recommended as coolants. A heavy base is used for threading, tapping or any coarse operations run at low spindle speeds, and high feeds. It is cut back with paraffin oil for forming, box turning, deep drilling, reaming, etc., where higher spindle speeds and low feeds are usually employed.

For comparison, the relative speeds of the principal machining operations are shown in relation to those employed on standard screw stock:

Operation	Type	Approx. Speed in Per Cent of Bessemer Screw Stock
Turning	410	65
	416	85
Forming	410	65
	416	85
Drilling	410	60
	416	80

FORGING:

Type 410 sections, if large, should be preheated and soaked at 1400-1600 Fahr. before raising to 2000-2200 Fahr. Otherwise, they may be heated slowly and uniformly to the forging temperatures. It is important that all sizes be held in this temperature range long enough to insure complete penetration. Heating above this range may lead to forging difficulties. Severe hot working below 1600 Fahr. is also avoided. Type 410, in common with all stainless steels, resists deformation more than carbon steels, consequently requires more hammer blows for reduction. Type 410 forgings are cooled slowly from the hammer as this grade is air hardening.

Type 416 may be forged or upset in the same manner as Type 410 except that it requires heating to approximately 2100 to 2300 Fahr. After thorough and uniform soaking it is worked on the high side of the forging range, finished above 1750 Fahr. and cooled slowly from the hammer.

WELDING:

Type 410 is readily welded. While rods or electrodes of similar analysis generally are used, those of Type 304 (18-8) are equally satisfactory. Type 410 is air hardening, and annealing after welding is necessary to restore full ductility. Small pieces may be annealed in their entirety, while the practical method for handling larger pieces is to torch-anneal and air cool.

As an added precaution, it is often desirable to preheat parts to be welded, thus minimizing the hazard of hardening cracks.

Type 416 is not generally welded.

BRAZING:

Brazing may be divided into two broad classifications: "Braze welding", wherein a bronze filler metal which melts in the temperature range of 1600-1700 Fahr. is used, and "low temperature brazing", employing silver brazing alloys (commonly called silver solders) which melt in the range of 1100-1600 Fahr. Braze welding is possible but is not generally recommended because of the danger of harmful intergranular penetration of the bronze alloy. Low temperature brazing is more frequently used. Those silver alloys which have a flow point at the lower end of the melting range are usually recommended because there is less danger of penetration and their color more closely matches that of the stainless steels.

SOLDERING:

Types 410 and 416 can be soldered using the common variety of solders such as half-and-half. The only requirements are sufficient heat and a flux that will "cut" the oxide film on the surface of the metal. Tinning is desirable but not essential.

Materials Work Sheet

RESISTANCE TO CORROSION

Type 410 is used in applications involving exposure to process solutions of neutral salts, basic salts, natural food acids, waste products, natural waters low in total solids, and even highly acid iron-bearing mine waters and copper solutions. It resists nitric acid fairly well; the sulphur acids moderately; the halogen acids poorly. To insure maximum corrosion resistance, parts may be "passivated" or "immunized" by immersing them for about 30 minutes in 20 to 40 per cent nitric acid at room temperature, washing in water and drying. In many instances, an addition of 1 per cent sodium dichromate further enhances the nitric acid solution.

The corrosion-resisting properties of Type 416 are comparable to those of Type 410, consequently the former may be used in most services where the latter is considered suitable.

GALVANIC CORROSION

Aluminum, zinc, cadmium and ordinary iron or steel all tend to protect the stainless steels and the attack, if any, goes to the other metal. Nickel, lead, copper, brass, graphite and silver, however, reverse the attack and by so doing are themselves protected. It is understood, of course, that galvanic couple is established between dissimilar metals only when they are in contact in the presence of a liquid which is capable of carrying electric current. The extent of the acceleration depends on many factors, including potential difference between the metals, conductivity of the electrolyte, and the relative areas exposed. When each of the two metals concerned is individually resistant to the corrosive condition, the attack generally is slight.

HEAT TREATMENTS

Quenching: Maximum hardness, in general, can be produced in sections 1-inch square by heating for 15 to 30 minutes at 1750-1800 Fahr. and quenching in oil. For small diameter wire, treated by the continuous method, 5 to 10 minutes at temperature is sufficient. Full hardness can be developed in small sections by air cooling. It is not advisable to water quench these steels when hardening, as cracking may result.

Stress Relieving: It is advisable to relieve stresses set up in the quenching operation by reheating to 400-800 Fahr. and air cooling. For small sizes, reheating for ½ to 1 hour is sufficient. For large sizes, up to 5 hours on the high side of the temperature range may be required. Such treatments have little effect on the hardness as quenched, but materially improve elastic properties and increase toughness.

Tempering: A wide range of physical properties can be developed by reheating the "as-rolled" or hardened material for 1 to 5 hours in the range of 1000-1400 Fahr. and cooling in air.

Annealing: Full annealing is accomplished by heating above the critical point—about 1550-1600 Fahr.—and furnace cooling. Depending on size, ½ to 5 hours at temperature is required. The rate of cooling should be carefully controlled, not faster than 50 degrees Fahr. per hour to 1200 Fahr., followed by air cooling. Hardnesses as low as 140-150 brinell are obtained in this manner. For intermediate processing operations, where maximum softness is not required, satisfactory tempers may be obtained by heating in the range of 1400-1450 Fahr., followed by air cooling. Hardnesses of 160-185 brinell usually result from this treatment.

Materials Work Sheet

DATA ON STOCK FORMS

Sheet Finishes

No. 1—Hot-rolled, annealed and pickled.
No. 2D—Full finish (dull).
No. 2B—Full finish (bright cold-rolled).
No. 4—Standard polish, one or both sides.
No. 6—Standard polish, Tampico brushed, one or both sides.
No. 7—High luster finish.
No. 8—Mirror finish.

Their General Applications

No. 1: For parts which require no further finishing—for oxidation or scale resistance.

No. 2D: Similar to No. 1, except slightly better finish due to cold rolling. For deep-drawn articles which are polished subsequent to fabrication.

No. 2B: Slightly harder temper than No. 1 and No. 2D, hence should not be used for exceptionally difficult deep-drawing work.

No. 4: For use where a polished finish is demanded. It is used for such equipment as dairy and ice cream tanks, soda fountains, meat packing and food equipment, etc.

No. 6: Tampico brushing of the No. 4 finish results in a satin finish with lower reflectivity than No. 4. For ornamentation where a brighter finish is undesirable.

No. 7: The buffing of No. 4 finish results in a sheet with a high degree of luster but in which some grit lines remain.

No. 8: The highest degree of polish commercially available. Grit lines and scratches removed. Suitable for press plates as used in the laminated paper and fibrous products industry. Also for mirrors and reflectors.

Standard Sheet
No. 1 and No. 2 Finishes
ASTM THICKNESS TOLERANCES

Specified Thickness (inches)	Tolerance Plus or Minus (inches)
.005	.001
.006 to .007	.0015
.008 to .016	.002
.0161 to .026	.003
.027 to .040	.004
.041 to .058	.005
.059 to .072	.006
.073 to .083	.007
.084 to .098	.008
.099 to .114	.009
.115 to .130	.010
.131 to .145	.012
.146 to .176	.014

Standard Sheet
No. 1 and No. 2 Finishes
ASTM WIDTH TOLERANCES

Specified Width (inches)	Tolerance (inches) Plus	Minus
Up to 42 inches	1/8	0
42 inches and over	1/8	0

LENGTH TOLERANCES

Specified Length (inches)	Tolerance (inches) Plus	Minus
Up to 120 inches	1/8	0
120 inches and over	1/8	0

Note: Sheet .131-inch and over in thickness, regardless of size, may have variations of plus or minus ¼-inch in width and length.

MATERIAL DESIGNATIONS

Federal
(410) QQ-S-763a Class 3 (bars and forgings, except for reforging)
(416) QQ-S-763a Class 6 (bars and forgings, except for reforging)

U. S. Navy
(410) 46S18 (INT) Class 3 (bars and forgings, except for reforging)
(416) 46S18 (INT) Class 6 (bars and forgings, except for reforging)
(410) 46S26 (INT) Class 3 (for reforging only)
(416) 46S26 (INT) Class 6 (for reforging only)
(410) 4720 (INT) Class 3 (plates, sheets, strips and structural shapes)

SAE
(410) 51210 (alloy steel) (416) x51410 (alloy steel)

Aeronautical Mat'l. Spec's.
(416) AMS 5610-A (bars, billets, forgings and shapes)

Cold-Rolled Strip
ASTM THICKNESS TOLERANCES

Specified Thickness (inches)	Tolerance—Plus or Minus (inches) 3/16 to ½-inch wide excl.	½ to 1-inch wide excl.	1 to 1½-inch wide excl.	1½ to 3-inch wide excl.	3 to 6 inches wide	Over 6 to 9 in. wide	Over 9 to 12 in. wide	Over 12 to 16 in. wide	Over 16 to 20 in. wide	Over 20 to 23½ in. wide
Under .006	.0005	.0005	.0005	.0005	.0005	.....	.....	.....	.....	.....
.006 to .009 incl.	.00075	.00075	.00075	.00075	.00075	.....	.....	.....	.....	.....
.010	.001	.001	.001	.001	.001	.001	.001	.001	.0015	.0015
.011	.001	.001	.001	.001	.001	.001	.001	.0015	.0015	.0015
.012	.001	.001	.001	.001	.001	.001	.0015	.0015	.0015	.0015
.013 to .014 incl.	.001	.001	.001	.001	.001	.0015	.0015	.0015	.002	.002
.015 to .016 incl.	.001	.001	.001	.001	.001	.0015	.0015	.0015	.002	.002
.017 to .019 incl.	.001	.001	.001	.001	.001	.0015	.0015	.002	.002	.002
.020 to .022 incl.	.001	.001	.001	.001	.0015	.002	.002	.002	.0025	.0025
.023 to .025 incl.	.001	.001	.001	.001	.0015	.002	.002	.002	.0025	.0025
.026 to .028 incl.	.001	.001	.0015	.0015	.0015	.002	.002	.002	.0025	.003
.029 to .031 incl.	.0015	.0015	.0015	.0015	.002	.0025	.0025	.0025	.003	.003
.032 to .034 incl.	.0015	.0015	.0015	.0015	.002	.0025	.0025	.0025	.003	.003
.035 to .039 incl.	.002	.002	.002	.002	.0025	.003	.003	.003	.003	.003
.040 to .049 incl.	.002	.002	.002	.002	.0025	.003	.003	.003	.004	.004
.050 to .068 incl.	.002	.002	.002	.002	.003	.003	.003	.003	.004	.004
.069 to .099 incl.	.002	.002	.002	.002	.003	.003	.003	.004	.004	.004
.100 to .160 incl.	.002	.002	.002	.002	.003	.004	.004	.004	.005	.005
.161 to .249 incl.	.002	.002	.003	.003	.004	.004	.004	.005	.006	.006

Note: Permissible variations in thickness are based on measurements taken ⅜-inch in from the edge on cold-rolled strip 1-inch or over in width and at any place on the strip on material less than 1-inch in width.

MATERIAL TRADENAMES

Type 410
Allegheny 12
Armco 13
Bethadur 410
Carpenter No. 1
Stainless Iron No. 12
Stainless "T"

Type "T"
Duro-Gloss C1
Lesco L
Midvaloy 13-00
Enduro S-1
Rustless 12
Uniloy 1409
USS 12

Type 416
Allegheny 12-EZ

Bethalon 416
Carpenter No. 5
Stainless Iron FM-2
F.M.S.

Type "FC"
Duro-Gloss FM
Lesco LS
Midvaloy 13-OOC
Enduro FC
Rustless 12 FM
Uniloy 1409-M
USS 12 FM

Producers
Allegheny Ludlum Steel Corp.
American Rolling Mill Co.
Bethlehem Steel Co.
Carpenter Steel Co.
Crucible Steel Co. of America
Colonial Steel Co. & Vanadium Alloys Steel Co.
Firth-Sterling Steel Co.
Jessop Steel Co.
Latrobe Steel Co.
Midvale Steel Co.
Republic Steel Corp.
Rustless Iron & Steel Corp.
Universal-Cyclops Steel Corp.
U. S. Steel Corp.

Correction

The value for Electrical Conductivity in the November, 1943, Work Sheet on types 302, 303 and 304 should be 2.4, not 24.

ASSETS to a BOOKCASE

Technique of Executive Control

By Erwin Haskell Schell, professor of business management, Massachusetts Institute of Technology; fifth edition; published by McGraw-Hill Book Co. Inc., New York; 252 pages, 5 by 7¼ inches, clothbound; available through MACHINE DESIGN, *$3.00 postpaid.*

With the success or failure of all human enterprise so intimately tied up with the quality of its leadership, it is surprising to find the prevalent lack of emphasis being placed on planned executive training.

Of course there is a vast difference between the teaching of factual sciences and a subject which is fraught with the intangible idiosyncrasies of human behavior. Then too, there are those who stoutly aver that "leaders are born", intimating that if one was not flecked by the gift-wand of destiny at birth, he had better relegate himself to anonymity and console himself as best he can.

But the more optimistic view—maintained by men such as Professor Schell—presupposes some of the qualities of leadership in all of us. Further, the contention is made that what qualities exist—whether of a limited or abundant nature—can, through proper training, be developed.

Just how the technique of the executive can be developed makes interesting reading. The author points out the importance of such qualities as enthusiasm, cheerfulness, calmness, stability, receptiveness, frankness, firmness, tolerance, patience, etc. He discusses the duties of the executive; how to overcome difficulties with subordinates, associates and superiors; how to obtain sincere, wholehearted collaboration. The chapter dealing with the executive's job in our national emergency is particularly timely.

□ □ □

Fundamentals of Vibration Study

By R. G. Manley, De Havilland Aircraft Co.; published by John Wiley & Sons, Inc., New York; 128 pages, 5⅜ by 8½ inches, clothbound; available through MACHINE DESIGN, *$2.75 postpaid.*

It is a recognized fact that maximum endurance and stability in mechanical structures cannot be achieved in the presence of inordinate vibration. Past tendencies to disregard vibration during the initial stages of design, thus leaving the consequent difficulties in the finished product up to vibration specialists, are now being superseded by the more logical procedure of taking vibration problems into account at the outset.

This book has been prepared in such manner as to make quickly available the kind of information which is required to solve practical problems in design.

Briefly discussed in its content are: Systems having one degree of freedom (undamped motion); damping and forced vibration; undamped motion with two degrees of freedom; many degrees of freedom; continuous systems (heavy shafts and beams), and complex vibrations. Its appendix offers a dissertation on pure and applied mathematics; frequency equations and numerical solutions; a variety of exercises with answers, and a bibliography.

□ □ □

Experimental Stress Analysis

Edited by C. Lipson, Chrysler Corp. and W. M. Murray, Massachusetts Institute of Technology; published by Addison-Wesley Press, Inc., Cambridge, Mass.; 156 pages, 8½ by 11 inches, clothbound; available through MACHINE DESIGN, *$3.00 postpaid.*

It would be difficult to overemphasize the profound effect which modern methods of stress analysis such as are covered in this book will have on machine design of the future. Endurance—a vital economic factor in all commercial mechanisms—is so greatly dependent on strength of design as well as strength of materials, that the growing significance of the new techniques in determining the true nature of stress and strain cannot fail to be recognized.

Where, as in many instances, mechanical structures are too complex to permit ready calculation of stresses, and where superimposed vibrational stresses are not susceptible to calculation, some form of "visual" analysis must be utilized.

Happily, new methods for determining stress concentrations, residual stresses, dynamically produced stresses, etc., are being developed constantly, to say nothing of the improvements being effected in established procedures. The day when lack of knowledge as to what a part would stand was camouflaged by the use of expensive, high-strength alloys, is fast on the wane. Today, spurred by more exacting economic requirements and bolstered by a more comprehensive ken of stress and strain, the designer more fully realizes that ordinary materials will suffice in most cases providing they are properly shaped and distributed.

For the mechanical engineer who wishes to keep abreast of the latest developments in the science of stress analysis, this book is recommended without reserve. A compilation of papers which were presented at the seventeenth Eastern Photoelasticity Conference held in Detroit last May, it offers a wealth of information on such phases of the subject as: Analysis with electric strain gages, analysis through use of brittle coatings, photoelasticity, etc. The many specific applications of the various techniques are discussed in comprehensible fashion, the written word being qualified by numerous photographs and drawings.

Design Abstracts

Improved Magnetic Alloys

DESIGN trend of electrical indicating instruments has been greatly influenced by the use of permanent magnet alloys developed during recent years. The ideal magnet from the standpoint of the instrument designer would be one having high coercive force, residual induction and available energy, and which had good machining and fabricating qualities. An approach to this ideal had been made in the cobalt-molybdenum-iron alloys commonly known as "comol". Comol, whose typical composition is 12 per cent cobalt, 17 per cent molybdenum, balance iron, contains a minimum of the critical metals; it can be cast easily, and when properly heat treated can be drilled, milled and machined. As a result, accurate machining dimensions permit the degree of precision which is required to utilize this material fully in instrument magnets.

A coercive force of about 245 is obtained as compared with 210 for 36 per cent cobalt. Residual induction is 10,300, higher than either 36 per cent cobalt or Alnico II, and maximum energy value is 1,100,000 as compared with 930,000 for 36 per cent cobalt and 1,650,000 for Alnico II.—*From a paper by M. S. Wilson and J. M. Whittenton, General Electric Co., presented at the A.I.E.E. winter technical meeting in New York.*

Electrical Systems for New Planes

THE simplicity, low maintenance, safety, and light weight of alternating-current motors and generators tends to favor the use of alternating current for large high-altitude aircraft. Commutation and brush wear problems are minimized, and the development of extremely high-speed motors appears more promising. Various desired utilization voltages can be obtained by transformers, which are lighter and simpler than direct-current voltage converters.

Although the 27-volt direct-current electrical system used in Flying Fortresses and Liberators is performing in a most satisfactory manner, the size, performance, and electrical load requirements of proposed new planes are such that the 27-volt direct-current system will not be adequate or efficient from the standpoint of weight, thus making it necessary to consider new types of systems.

In view of the work that has been and is being done, the aircraft industry will have at its disposal electrical systems at several voltage levels, depending on the amount of power used and transmission distances. These include 6 or 12-volt direct-current systems for very small or private planes; 27-volt direct-current systems for small and medium-size planes; 27/120-volt direct-current systems for large, medium-altitude cargo or transport planes; and 120/208-volt 400-cycle alternating-current systems for large, high-altitude long-range planes.

The generators of both alternating-current and direct-current aircraft systems usually are driven from main engine accessory power take-off shafts. Considerable development work is now in progress on variable-ratio constant-speed transmissions and governors to provide a constant-speed drive for the generators. Fully automatic synchronizing equipment for 400-cycle alternating-current aircraft systems already has been designed.

Selection of 120/208-volt, 400-cycle, 3-phase as a voltage and frequency level for alternating-current systems in aircraft has proved a particularly happy choice in that this voltage is high enough to transmit the required amounts of electric power economically over the distances required by the largest planes, and is high enough for the largest amounts of auxiliary electrical power in any planes now being designed or likely to appear in the near future. The voltage is low enough so that no unusual difficulties are experienced in switching, even at very high altitudes.—*From a paper by W. K. Boice and L. G. Levoy, General Electric Co., presented at the recent annual meeting of the A.I.E.E. in New York.*

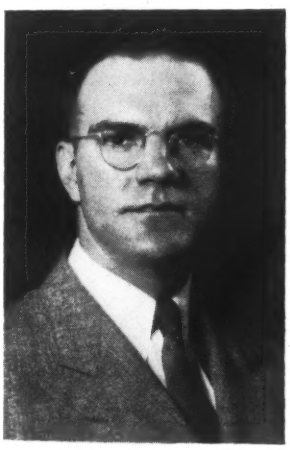
The Plastics Outlook

PLASTICS will never replace metal for most uses where metals were employed prior to the war, nor will the future lead to devastating competition from new low-priced availability of the white metals such as aluminum and magnesium. Sound engineering principles and good business practices will ultimately result in each of the metals and each of the plastics finding its own sphere of activity. That company with the most active and reliable imagination will do the best job of utilizing the increased availability of plastics and finding profitable postwar civilian uses for the new materials.

Total tonnage of the plastics industry is still exceedingly small as compared with the output of steel and other metals, but it has been estimated that approximately 550,000,000 pounds of plastics and synthetic resins were produced during 1943. This does not include the alkyd resins or other plastics employed for protective coating purposes. Approximately 80 per cent of this entire amount was used for direct and indirect military purposes, with the other 20 per cent devoted to highly essential industrial and civilian needs. It is obvious, therefore, that the plastics industry is producing for war at the rate of approximately 440,000,000 pounds per year, and the vast majority of this material will become available for civilian uses the minute the last gun is fired.—*From a paper by Clinton Rector, chief of the plastics section, chemicals branch of W.P.B., presented at the recent Pacific Coast section plastics conference in Los Angeles.*

Men of Machines

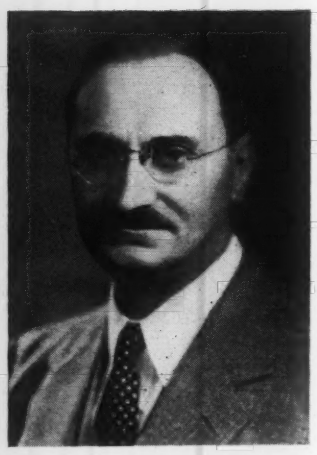
PREVIOUSLY assistant chief engineer, John W. Kelly has been appointed chief engineer of Adel Precision Products Corp., Burbank, Calif. Mr. Kelly, a specialist in hydraulic and hydro-electric actuating mechanisms for aircraft, armament and general industrial application, is responsible for many changes made by the company in its hydraulic systems. Prior to becoming connected with Adel Precision Products Corp., he was a lead man in charge of power plant installations at Vultee Aircraft Inc. He previously had spent three years with Douglas Aircraft Co. as power plant and hydraulics engineer. Mr. Kelly has been issued patents on several of the individual control devices currently manufactured by the Adel corporation. His background in the hydraulics field will be of considerable assistance to him in fulfilling the various duties of his new position as chief engineer of Adel which organization he joined four years ago.



AMONG OTHER recent appointments by the Pesco Products Co., is that of Jay M. Roth as director of engineering for pump and automotive developments. Mr. Roth is a mechanical engineer, graduating in 1931 from Ohio State university where he received the Lamme Scholarship award. Upon graduating he started his career at the Air Corps Power Plant Branch at Wright Field as a test engineer. After obtaining some experience on engines he was made project engineer and assigned to the development of engine-driven pumps for operating gyroscopic flight instruments. Mr. Roth left the Materiel Command in 1937 to join the engineering staff of Pump Engineering Service Corp., predecessor of the Pesco company. He held the position of assistant to the chief engineer until 1938 when he was made assistant chief engineer. From this post he was promoted to chief engineer in 1940 and served in that capacity until his present appointment.

AN EXECUTIVE and design engineer in the field of aeronautical engineering and diesel engineering, Martin J. Berlyn of Montreal is the new vice president in charge of engineering of American Bosch Corp. Mr. Berlyn has been connected with the Dominion Engineering Works Ltd. of Montreal for the past fifteen years, most recently in the capacity of manager and chief engineer of the diesel engine division. He also has been identified with the aviation industry, and only recently has terminated his service with the Royal Canadian Air Force, where as a Squadron Leader and chief technical instructor of the School of Aeronautical Engineering he has been in close touch with the engineering problems of military aviation. Mr. Berlyn was born in England in 1902, and attended the King Edward School at Birmingham before entering Trinity Hall at Cambridge University from which he obtained his degree. He then became engaged in research work in magnetos and automotive ignition systems with the British Lighting & Ignition Co. Subsequently he was

connected with Armstrong-Whitworth Aircraft, Coventry, England; Canadian-Vickers, Aircraft Division, Montreal; and Curtiss-Reid, Montreal. In his new position with American Bosch Mr. Berlyn will have full charge of design and development work on fuel injection, ignition and other equipment of the company.



NOMINEE FOR the office of president of the American Institute of Electrical Engineers is Charles Alfred Powel, manager of the headquarters engineering departments of Westinghouse Electric & Mfg. Co. Born in France in 1884, he was educated in England and Switzerland, and was graduated in 1905 from the Institute of Technology, Bern, Switzerland. In the same year he became associated with Brown, Boveri & Co., Baden, Switzerland, in the applications engineering department. As resident engineer for that company, he was sent to Japan in 1911 and four years later returned to England to the civil branch of the British Ordnance department. From 1916 to 1918 he was engaged in the inspection service of the British War Mission in the United States. A year later he became connected with the central station section of general engineering department of Westinghouse Electric & Mfg. Co., and was made manager of this department in 1935. He became manager of the industry engineering department in 1938 and in two years, manager of the headquarters engineering department—the position he now holds. Mr. Powel joined the institute in 1920 and in 1941 became a Fellow. He served as a director from 1936 to 1940, and as a vice president from 1942 to 1943. A past chairman of the Pittsburgh section, he has served as a member of many of the Institute's national general and technical committees such as power generation, marine transportation, electric welding, electrochemistry and electrometallurgy, education, standards, etc.

FRANK R. BENEDICT, who joined Westinghouse Electric & Mfg. Co. in 1928, has been appointed manager of product performance analysis. Mr. Benedict helped develop microscopes that enable engineers to check the ignition "heartbeats" of giant aircraft engines.

♦

DELMAR ANDERSON, superintendent in charge of production planning and tool design for the Buffalo warplane plants of Curtiss-Wright Corp., recently was elected president of the National Society of Plastic Tooling.

♦

HENRY B. BRYANS has been made president of American Standards association. GEORGE S. CASE, chairman of Lamson & Sessions Co., was re-elected vice president. H. S. OSBORNE, chief engineer, American Telephone & Telegraph Co., and E. C. CRITTENDEN, assistant director of National Bureau of Standards, are standards council chairman and standard council vice chairman, respectively.

♦

C. T. MORSE has been elected president for 1944 of the National Association of Fan Manufacturers, and J. M. FRANK of Ilg Electric Ventilating Co., is vice president of the association.

♦

EARLE S. TOMKINSON has been advanced to the position of designer at Caterpillar Tractor Co., Peoria, Ill.

♦

C. W. VAN RANST, automobile and engine designer, has joined F. L. Jacobs Co. where he will be responsible for the company's defense and postwar development program. In his previous connection with the Ford Motor Co., he was associated with the design of aircraft engines which later were converted for use in Army tanks. Upon completion of work on the tank engine, Mr. Van Ranst came to the Jacobs company.

♦

ROBERT A. HITCH, previously associated with Glenn L. Martin Co. doing power plant development work, has joined General Electric Co., Lynn, Mass., as engineer.

♦

VERNON L. DURNSTEIN, formerly assistant chief engineer at Buda Co., has been made chief engineer in the diesel engineering division of Harnischfeger Corp., Port Washington, Wis.

♦

FRANK W. RIDDELL, former engineer at Lycoming Division, Aviation Corp., is now designer at Pratt & Whitney Aircraft Corp., East Hartford, Conn.

♦

EARLE S. MACPHERSON has become assistant engineer, head of Army contact at Chevrolet Motors division, General Motors Corp., Detroit. He moves up from the position of engineer in charge of truck and passenger car design.

♦

CHESTER ALAN FULTON has been elected president of the American Institute of Mining and Metallurgical Engineers. Mr. Fulton is president of Southern Phosphate Corp., Baltimore.

JOHNSON BRONZE

SLEEVE TYPE BEARINGS

For your POSTWAR Product

Babbitt Lined BEARINGS

SLEEVE TYPE BEARINGS

Cast Bronze Bearings
Cast Bronze Graphited
Sheet Bronze Bearings
Sheet Bronze Graphited
Bronze and Babbitt Bearings
Steel and Babbitt Bearings
Steel and Bronze Bearings
Ledaloyl
Self-Lubricating Bearings
Electric Motor Bearings
Automotive Bearings
Bronze Bars
Bronze Castings

Any Type
Any Size
Any Quantity

Designers of the postwar automobile may worry about the lines of their new car . . . or synthetic rubber versus natural . . . or the thousand and one new developments that have come as a result of the present conflict. There is, however, one worry that they can forget . . . and that's the type of bearings to use in the engine. Years of research and practical tests have definitely proven that babbitt lined bearings deliver the best all around performance for internal combustion engines.

The dual construction of this type of SLEEVE BEARING provides conformability to the shaft . . . resistance to pounding . . . smooth, quiet performance at high speed . . . exceptionally long bearing life, and excellent running-in properties.

Babbitt Lined Bearings are ideal for a wide variety of applications. They are "tailor made" in the sense that you have a wide range of alloys to select from in the babbitt, and a choice of backing material—either bronze in any of the various alloys, or steel. Why not permit a Johnson Engineer to study your bearing problems? He can easily advise you on the *correct* type of bearing for each application. There is one located in the cities listed below—ready to serve you.

DISTRICT SALES OFFICES: Atlanta · Boston · Buffalo · Chicago · Cincinnati · Cleveland · Dallas
Detroit · Kansas City · Los Angeles · Minneapolis · New Castle · New York · Newark · Philadelphia
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SLEEVE BEARING HEADQUARTERS
525 S. MILL STREET NEW CASTLE, PA.

The MOST COMPLETE SLEEVE BEARING SERVICE *in the* WORLD

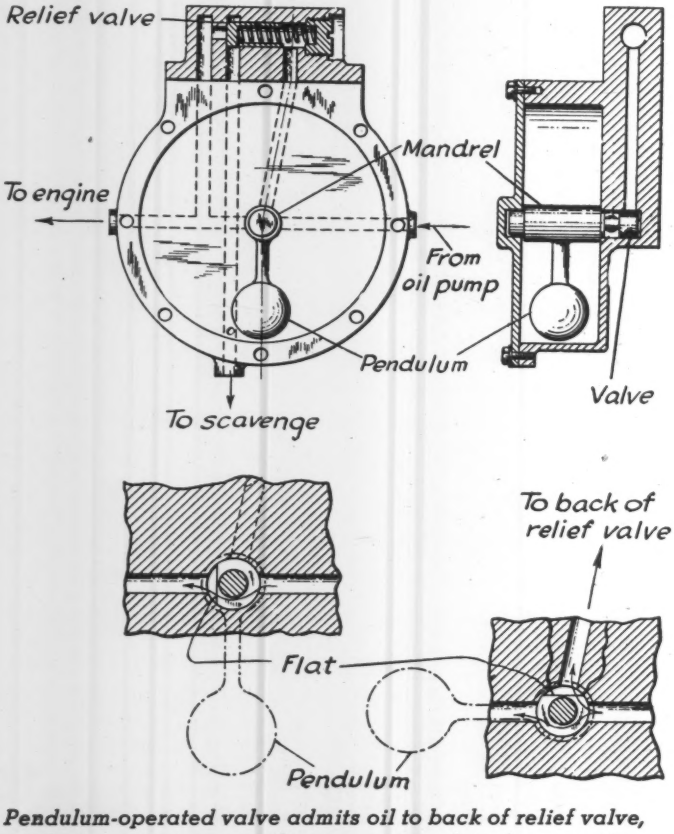
Noteworthy Patents

Controlling Engine Oil Flow

WHEN an airplane is subjected to a dive, the engine tends to overspeed. During such overspeed it needs a greater supply of oil for proper lubrication than ordinarily is available due to the fixed peak pressure to which the lubricating system is limited by the pressure-relief valve. Patent 2,336,784 recently assigned to Wright Aeronautical Corp. covers means for augmenting the oil pressure fed to the engine parts at such times as the engine tends to overspeed, normally whenever the aircraft assumes a nose-down position.

Shown in the accompanying illustration, the device consists of a special valve controlled by a pendulum which influences the operation of the pressure relief valve. The axis of the mandrel which supports the pendulum is disposed parallel to the transverse axis of the aircraft so that the pendulum, under uniform flight conditions, will hang toward the earth regardless of the nose-up or nose-down attitude of the airplane. The mandrel carries a necked-down portion and a valve portion projecting into a housing having a fore-and-aft drilling aligned with the necked-down portion. This drilling forms part of the main supply line from the oil pump to the engine bearings. The necked-down portion offers no obstruction to the free flow of lubricant from the pump to the engine at any time.

Upper part of the housing contains a spring-loaded pressure-relief valve of more or less conventional construction, which opens in case the oil pressure delivered by the pump exceeds the pressure furnished by the spring, allowing surplus oil to return to the engine pump and limiting engine oil pressure to a maximum value. The pendulum-operated valve is provided with a flat which permits the passage of pressure lubricant through another drilling to the space behind the relief valve. This occurs when the relative position of the pendulum and housing is as shown in the lower right-hand illustration and takes place when the airplane assumes a nose-down attitude. In this position oil pressure behind the relief valve substantially balances the pressure on the other side and the valve is held closed by the spring regardless of the actual pressure existing. Because the oil pump capacity exceeds the requirements of the engine, pressure in the system rises, causing an increase in the flow of lubricant to the engine. Whenever the plane assumes a normal attitude, the relative pendulum position is as shown in the lower left-hand illustration. The pressure oil behind the relief valve then bleeds off, allowing it to function in its normal manner to limit maximum oil pressure.



Pendulum-operated valve admits oil to back of relief valve, rendering it temporarily inoperative and permitting pressure rise and increased oil flow

Correcting Aerodynamic Unbalance

IT HAS been found that aircraft propellers which are perfectly balanced both statically and dynamically at low altitudes may become a source of vibration when operated on an airplane at high altitude. This is due to the fact that, although an aerodynamic defect can be balanced out by a redistribution of mass for any one altitude, when the air density is changed the aerodynamic forces vary in proportion while the mass distribution remains constant. In order to overcome this difficulty the propeller must be balanced initially for both mass and aerodynamic effects. Because both effects occur at the same time it is necessary to provide means for separately indicating the defects. Such means are covered by patent 2,338,293, recently assigned to United Aircraft Corp.

Principal feature of the design is a shroud or housing which completely encloses the propeller while the latter is mounted on the balancing machine. The shroud is supported on a shaft which is coaxial with the propeller shaft and which can be rotated by a separate motor. Thus when the propeller is enclosed in the shroud and the shroud is rotated with the propeller at approximately the same speed, the aerodynamic effects on the propeller are eliminated and it may be considered that vibrational effects are due to mass unbalance. After the mass unbalance has been corrected the shroud is removed and the propeller again tested. This time, vibrational effects may be assumed to be due to aerodynamic unbalance, which may then be corrected.

wherever a tube is used...
INDUCTION HEATING
Electronic heaters with inductor coils connected to terminals of Oscillator Type Tubes, are used for brazing, soldering, hardening, and bonding of small diameter parts, thin sections and low resistance materials.

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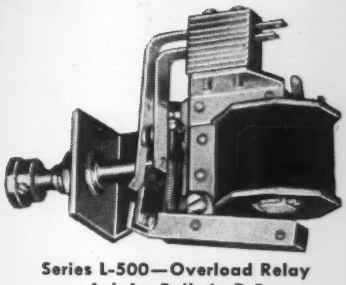
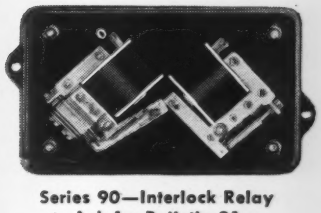
Induction heating costs less, is more dependable and easier by the electronic method. Safer, too, even with inexperienced hands, when protected from improper operation by quality relays such as the Guardian Series 90 Interlock Relay and the Series L-500 Overload Relay.

Series 90 Interlock Relay locks mechanically from an electrical impulse and unlatches or resets from another such impulse. Combinations up to DPDT available with 12½ amp. contacts rated at 110 v., 60 cycles, non-inductive load. Coil resistances up to 10,000 ohms. For AC or DC operation.

Series L-500 Overload Relay, manual reset, protects DC circuits against abnormal current surges where current conditions are constant. Contacts can take severe overloads undamaged, cannot be reset during overload. Rated for 1500 watts on 110 v., 60 cycle, non-inductive AC.



Series L-500—Overload Relay
Ask for Bulletin R-5



Series 90—Interlock Relay
Ask for Bulletin 21

Consult Guardian wherever a tube is used—*however*—Relays by Guardian are NOT limited to tube applications but may be used wherever automatic control is desired for making, breaking, or changing the characteristics of electrical circuits.

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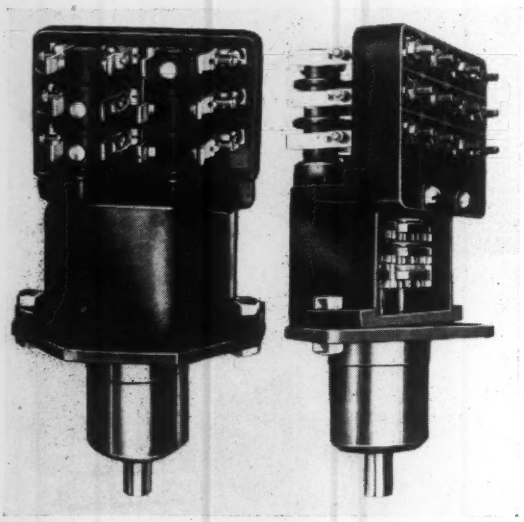
New PARTS AND MATERIALS

Geared Limit Switch

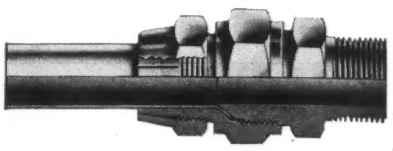
AVAILABLE ON short delivery on orders bearing suitable priorities, the geared limit switch offered by Philadelphia Gear Works, Erie avenue and G streets, Philadelphia 34, incorporates two individually adjustable trains of intermittent gears. Each train operates one contact drum, providing three double-break contacts. Of these, two contacts may be normally open and one normally closed or vice versa. The switch can be adjusted for operation between 2 and 3333 turns of driving shaft at speeds ranging from 10 to 500 revolutions per minute. Contacts are double-break, self-cleaning due to wiping action, and of silver, rated for voltage up to 550 volts alternating or direct-current. Of molded construction, the switch is for



applications requiring a large number of turns between limits, such as for the control or the position indication of hoist or crane mechanisms, automatic doors, valves or dampers, or any mechanism where precision control or position indication is required. Mounting can be in any position and watertight weatherproof enclosures can be supplied, if desired.

Tube Fittings Seal and Grip

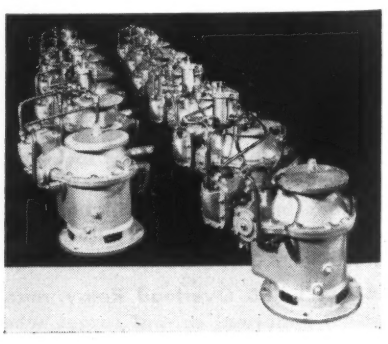
PROVIDING LEAKPROOF, vibrationproof connections without the necessity of threading, welding or soldering of tubing, a new and improved type of tube fitting has been announced by Cotner-Wilkinson Co., division of Logansport Machine Co. Inc., Logansport, Ind. Distinctive features of the new "Collet Grip" fittings include a saw slot in the collet which permits a firm grip of the surface of the tubing, eliminating transmission of shock or vibration to the flare seal; a compression nut which compresses the collet nut to the tube, giving it a firm grip that does not loosen under vibration or shock from the hydraulic system; and the long bearing surface of the collet nut, which grips and holds the tube securely without damage by cutting or digging. The fittings provide connections that,



according to the company, will hold beyond the strength of the tubing itself and may be used repeatedly. Easily assembled or disassembled, the fittings are made in both double and single-nut design. The double nut type is available in five standard shapes for tube sizes from ¼-inch to 1 inch, while the heavy duty fittings (double nut) are made in the same standard shapes for tube sizes ranging from ½-inch to 2 inches. Single-nut fittings are simplified for close work and are obtainable also in five standard shapes for tube sizes from ¼-inch to 1 inch. Special shapes and sizes can be furnished to specifications.

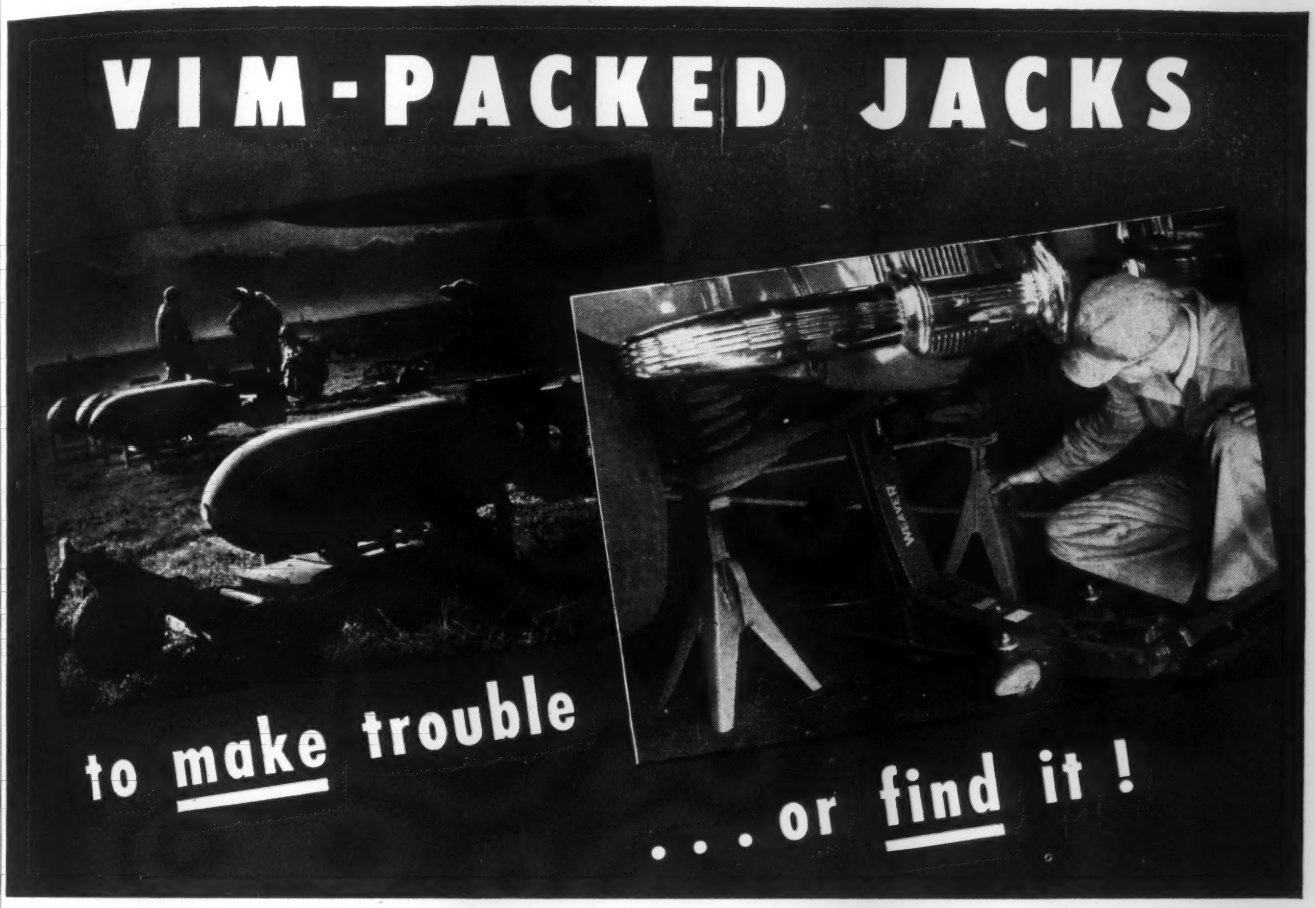
Turbine Reduction Units

SELF-CONTAINED, industrial turbine reduction units, announced by the Cone-Drive division, Michigan Tool Co., 7171 East McNichols road, Detroit 12, have been developed from the naval turbine reduction units produced by this company for the United States Navy. Built around the use of area-contact, double enveloping Cone-Drive



gearing, they are compact and are available in 26½ and 37-horsepower ratings. The former is the output rating of a 4½-inch center-distance unit, while the larger one has gears of only 6 inches center distance. Both types are designed to operate on input speeds of 5000-6000 revolutions per minute, and are available in a selection of standard ratios ranging from 3½ to 8:1. They are com-

VIM-PACKED JACKS



to make trouble . . . or find it!

You'll find hydraulics everywhere, in war and in peace. Note the above contrasting uses—the Weaver hoist on which the special delivery to Berlin rests so safely prior to being lifted into bomb racks, and the automotive jack used to find trouble and repair it, as pictured at the right.

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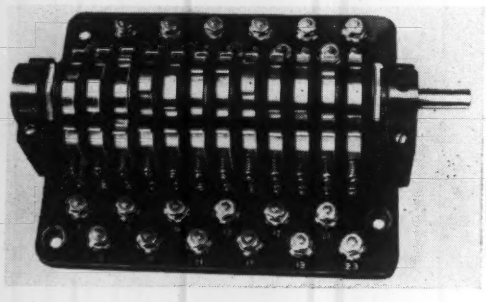
pletely equipped with built-in oil pump and oil cooler, oil filter and pressure relief valve, pressure gage, thermometer and oil level gage. Lubrication system in the units is of the force-feed type.

Rotary Type Limit Switch

FOR APPLICATIONS requiring a multicontact limit switch having flexibility and ease of adjustment, the new rotary type switch announced by Philadelphia Gear Works, Erie avenue and G streets, Philadelphia 34, has proved suitable. This twelve-circuit, direct-connected, rotary type switch has twelve individual contact drums, each separ-



ately adjustable by small increments. It can be used either as a continuously rotating or reversing switch, for operation in any position. Its all molded construction provides accessible terminals, silver-to-silver wiping and self-cleaning, double-break contacts, and needle bearing shaft suspension. Contacts are rated for 125 volts alternating current applications.

Constant Level Lubricator

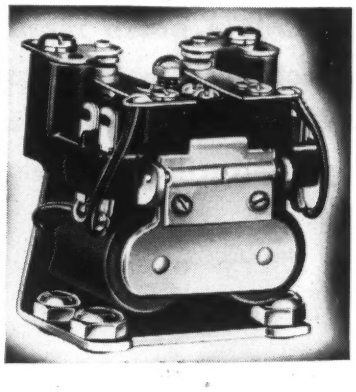
OFFERING AN automatically controlled visible oiling method for most types of bearings, the Type J constant level lubricator recently improved is now being marketed by Oil-Rite Corp., 3476 South Thirteenth street, Milwaukee 7. Insuring lubrication automatically as needed, the lubricator can be used on electric motors, pillow blocks, textile and paper-making machinery, pumps, air-conditioning equipment, ventilating fans, gearboxes, and wherever bearings and other machine parts must be provided with an ample supply of lubrication. When reservoir is filled with oil, enough oil will be released to bring the level in bearing oilwell to the oil level line marked on the base of the lubricator. When bearing oil level falls, air is admitted into the reservoir through a vent, automatically releasing enough oil to restore the original level. The lubricator consists of three parts: The base, dome and reservoir. In the base are a bottom and side outlet for connection to bearings, either or both of which may be used. In the center is a large intake opening which may be connected to a filter or to a breather tube on the bearing. The reservoir is sealed to a collar with a plastic cement which provides a strong bond and is not affected by oil, water, heat or acid. The lubricators are available in four standard sizes with capacities of 2, 4, 8 and 16 ounces.

Spring-Grade Beryllium Copper

BERYLLIUM COPPER wire is being produced by Little Falls Alloys Inc., 189 Caldwell avenue, Paterson, N. J., in a spring grade. A new silver coating process is used which reduces wear on coiling tools, increases surface electrical conductivity, reduces surface attack during hardening, and makes the wire easy to identify at a glance. Minimum tensile strength is 185,000 pounds after hardening. This wire, trademarked "Silvercote", is now available in sizes .064 to .007-inch. Previously obtainable sizes were .0359 to .010-inch. Sizes .0201 and under are supplied in spools, in average weights of 10 pounds.

Small Dual Coil Relay

DEVELOPED BY Allied Control Co. Inc., 2 East End avenue, New York 21, to meet the demand for a small power relay with sensitive (½ watt) operation, a new dual coil unit, Model BOY, has been announced. Through

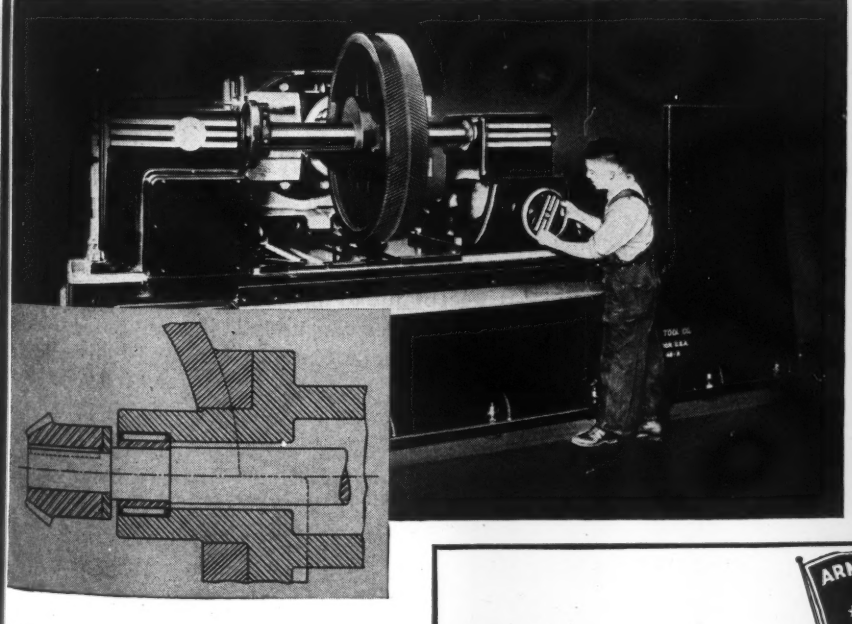


double-coil construction, power consumption required has been reduced, making the unit suitable for applications where current is insufficient to operate a single-coil power relay. The two coils of the BOY relay may be connected in series for operation at one voltage and in parallel for a second voltage; or one coil can be for operation while the other is used for holding. Coils are cellulose-acetate sealed to withstand salt spray and high humidity. The relay features semibalanced armature construction and is ruggedly built to withstand 10g. Dimensions are 1⅝

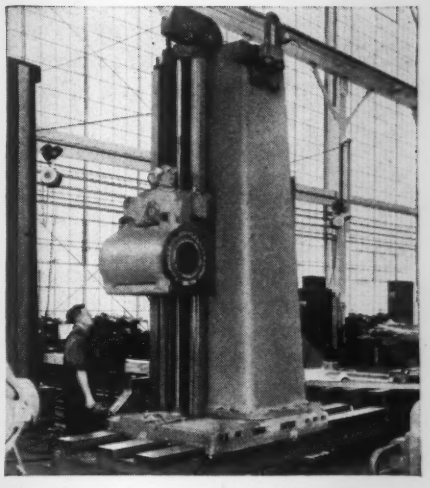
(*Continued on Page* 198)

IN THE NEWS
WITH TORRINGTON BEARINGS

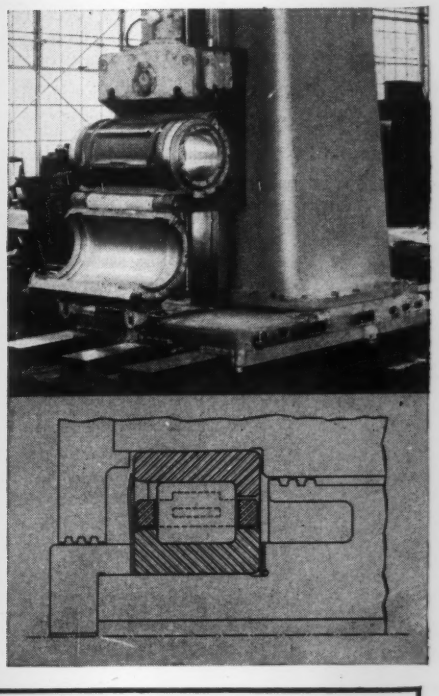
TANK BUSTERS must pack the power not only in armament but for locomotion as well. The M-12 shown here which mounts a 155 MM. field gun and designed to have all the speed and mobility of a tank, was among the first of our Army's fleet of tank destroyers. In the compact, efficient power plants of these modern weapons are small, heavy duty Torrington Needle Bearings—used to carry and transmit the heavy loads under the toughest imaginable service conditions. For applications such as automotive transmissions, universals and differentials, the high unit capacity, small size and efficient lubrication of Needle Bearings make them ideal for all radial load applications.



GEARS FOR TURBINES up to 4 feet in diameter and 20-inch face width are processed on this Michigan Tool Company's gear finishing machine. Several key points in the infeed mechanism are mounted on Torrington Needle Bearings to insure smooth, friction-less operation. Typical is the application of a Torrington Type DC Needle Bearing (with hardened inner race) on the hand operated infeed shaft shown in the cross-section.



SUPPORT FOR A 24-FOOT BORING BAR is the job of this large end support bearing for a floor type horizontal boring, drilling and milling machine photographed in the shops of Giddings & Lewis Machine Tool Company. Supporting this huge shaft in the housing shown open in the close-up photo (below) are two special 18-inch O. D. radial roller bearings supplied by Torrington's Bantam Bearings Division. These bearings illustrated in the cross-section illustrate the type of service Torrington is equipped to render in the design and manufacture of special bearings to meet the unusual and out-of-the-ordinary bearing requirements. Why not TURN TO TORRINGTON for specialized assistance on your next bearing problem?



ARMY E NAVY

TORRINGTON BEARINGS

STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

THE TORRINGTON COMPANY · BANTAM BEARINGS DIVISION

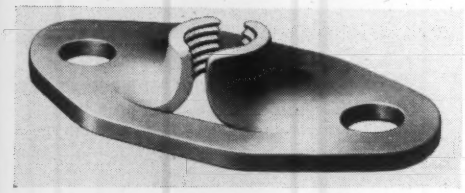
SOUTH BEND 21, INDIANA

(*Continued from Page* 192)

x 1⅞ x 1½ inches. Weight is five ounces. In design the relay meets Army, Navy and CAA specifications.

High-Stress, Lightweight Nuts

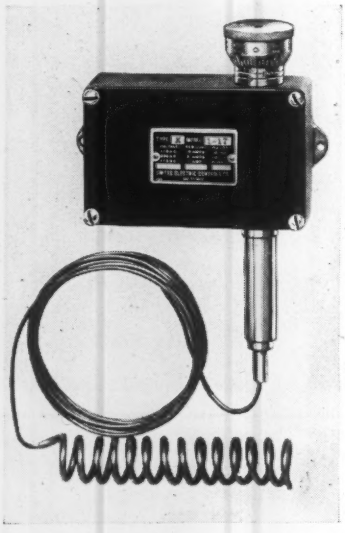
CONFORMING TO AAF specification No. 25531, a new "Hi-Stress" speed nut (identified as No. A6103H-1032) has been developed by Tinnerman Products Inc., 2085 Fulton road, Cleveland 13, and approved by the Army Air Forces. Light in weight, the new nut is a one-piece integral unit



with a low installation torque that permits quick insertion of screws and bolts. It is interchangeable with nut plate AN362, for high-temperature applications. The nut has another feature—that of retaining its self-locking torque even after many removals under service conditions.

Thermostat Has Wide Range Control

FOR TEMPERATURE CONTROL between —120 degrees and 600 degrees Fahr., United Electric Controls Co., 69-71 A street, Boston 27, has introduced its Type K thermostat. Various models of this precision-built thermostat cover calibrated ranges up to 550 degrees Fahr., and the units have a differential operation of from .5 degrees



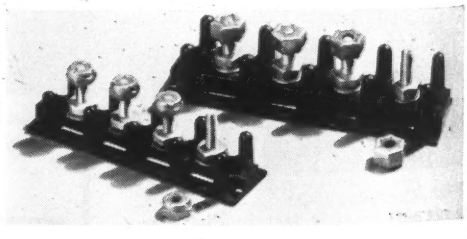
Fahr. to 1 degree Fahr. in liquids, sub-zero testing cabinets and mechanical convection ovens, to 2 degrees Fahr. in gravity type ovens. Containing no delicate mechanisms, the thermostat can withstand vibration, shock and moisture conditions. Designed to control over a wide range or any intermediate range, provision is made in the form of a simple calibration correction so that control performance is as though the control were furnished for the specific range desired. Effective ambient temperature compensation permits use of a precise wide-range temperature responsive device, the liquid-filled bellows, bulb and tube assembly. Advantages of this assembly are quick response and uniform bellows motion per degree. Application of this motion to a sensitive snap-acting switch produces close differential equal at all temperatures. The micrometer adjustment being precise, having the same movement per degree, covers a uniform amount of range for each complete turn.

High-Strength, Heat-Treated Glass

HEAT-TREATED GLASS for use in dehydrating equipment has been introduced by The Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, to take the place of wartime critical methods. The new glass is four times as strong, flexible and tough as ordinary glass of equal thickness, making it more than adequate for any dehydration applications. It withstands constant temperatures up to 650 degrees Fahr., as well as sudden variations of temperature necessary in certain processes. Being sanitary and noncorrosive, the glass is immune to attacks of practically all food acids, fumes and chemicals. Available in either opaque or transparent form, the material is made in a variety of shapes and forms to fit requirements of different types of food processing equipment. It can also be obtained in bent shapes, in pierced disks, narrow strips, flat plates for tank linings, etc., and comes in a wide range of thicknesses.

Connector Panel Assemblies

MOLDED CONNECTOR panel assemblies are now being supplied by Plastic Manufacturers Inc., Stamford, Conn., as complete units ready for installation. Use of these assemblies eliminates the necessity of stocking seven different items and providing for their assembly. All component parts are in accordance with nationally recognized standards, acceptable to manufacturers of Army and Navy aircraft. Because of the adaptability of the assemblies to general use, manufacturers of ground and sea-going



equipment have adopted them as well. The units are available in P1, P3-2 and P3-3 types. Both the P1 and P3 can be ordered in varying lengths to suit requirements as to the number of posts needed and the available room in junction box assemblies. Variations can be scheduled

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where quantities merit such modification. According to the company, use of these connector panel assemblies has been so extended that the National Aircraft Standards committee has been studying the possibility of assigning national numbers to cover them.

Heat and Oil-Resistant Rubber

FOR USE IN oil seals, gaskets, "O" rings, and other types of fuel seals and parts requiring resistance to heat and oil, Los Angeles Standard Rubber Inc., 1500 Gage avenue, Los Angeles, has developed a new general-purpose rubber, Buna N stock (143-18). In addition to its high heat and oil-resistant qualities, it remains flexible to 35 degrees Fahr. below zero.

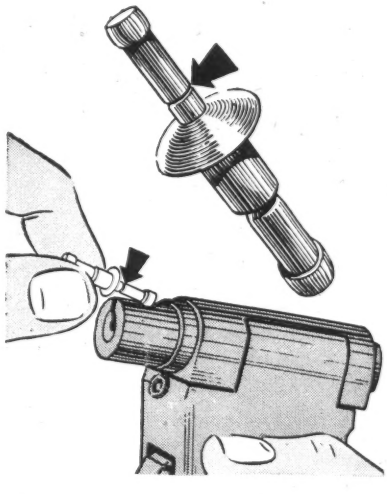
Remote Control Right-Angle Adapter

FOR CLOSE CLEARANCE connections in flexible shaft control, F. W. Stewart Mfg. Corp., 4311 Ravenswood avenue, Chicago, has perfected its remote control right-angle adapter. It is designed to provide a connection for flexible shaft control where space limitations do not permit a direct conjunction without bending the shaft in too small

a radius. Being light in weight it is of primary value to the aircraft industry, although it has been found applicable to many other fields where the power transmission must of necessity be at a small radius. Size of the unit is 2 x 1⅞ x ¾ inches; weight only .141-pound. Gear ratio is 1 to 1. Maximum torque is 25 inch-pounds. Connections are ⅝ x 24 NF 3 threads. While the adapter is light in weight it is sturdy and simple in construction.

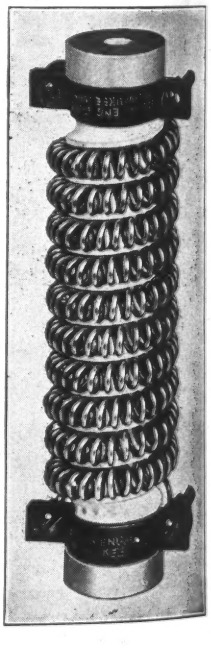
Blind Rivets Are Notched

BLIND RIVETS produced by Cherry Rivet Co., Los Angeles 13, are now being supplied with a new feature—that of a notch in the pulling mandril of the self-plugging type rivet—which provides several advantages. The notch serves to inhibit the flow of metal in the upsetting process employed in forming the pulling head. As a result the rivets are now more uniform. Another advantage is that a shorter mandril is required, effecting a saving in material as well as providing for a shorter drawbolt movement, making possible the use of the gun for longer grip lengths. Designed to speed production by making easier to handle riveting jobs in difficult places and where there is access to only one side of the work, the

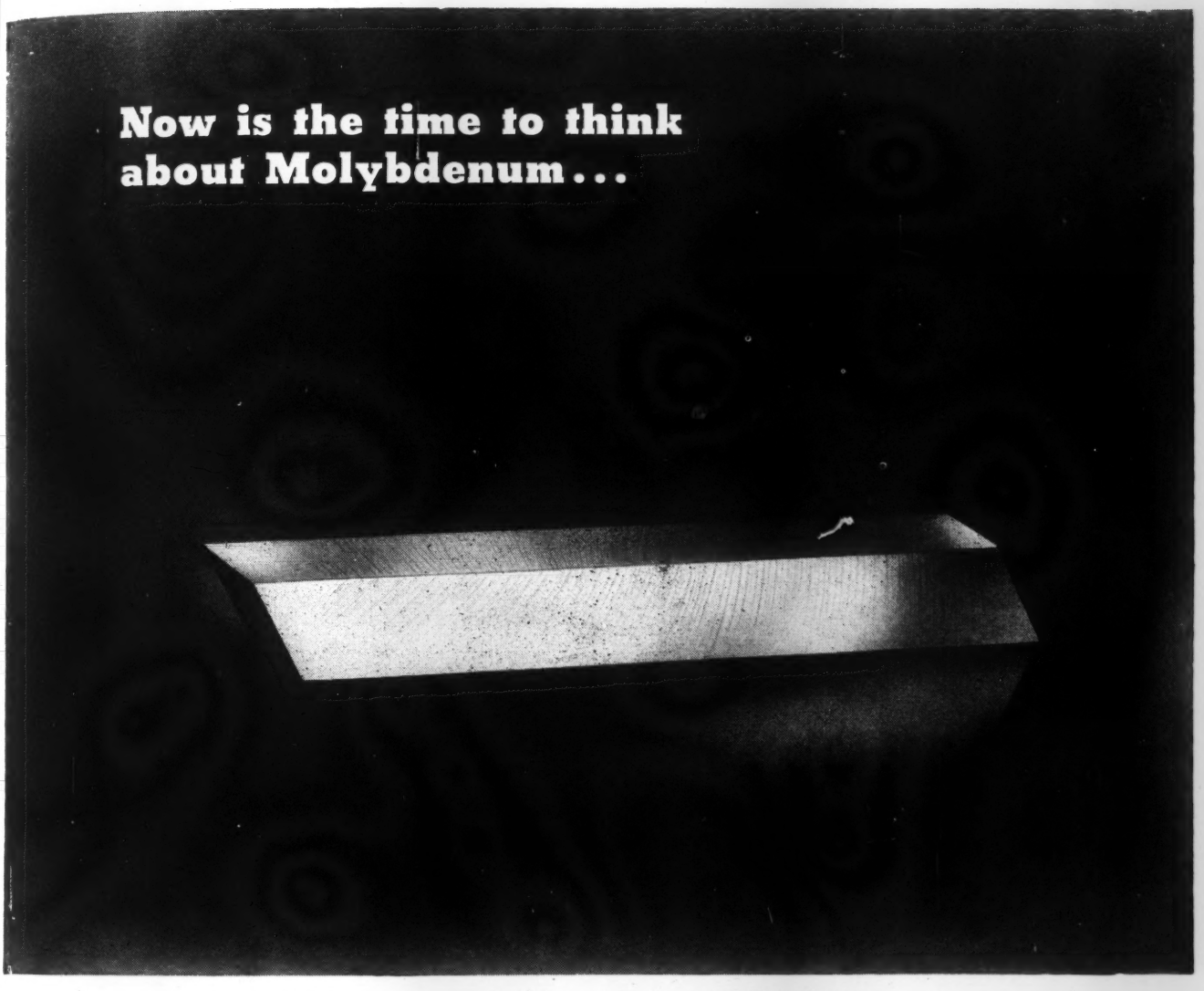


rivets are headed-up by a hand gun or power gun. The gun is used to pull the mandril of the rivet with sufficient force to head the rivet on the blind side and break the stem at the notched portion.

Radiant Heater and Resistor



A DEVELOPMENT of Techtmann Industries Inc., 828 North Broadway, Milwaukee—the new Regan radiant heater and resistor has been designed for use in annealing and drying ovens, electric furnaces and wherever high-temperature radiant heat is required. It can be installed in air ducts, and is available in any wattage or voltage. The element consists of a helical coil of nickel-chromium wire wound spirally around a refractory core. The element can also be furnished in copper-nickel alloy which will not oxidize due to ordinary moisture conditions but which cannot be operated over 600 degrees Fahr. Where high operating temperatures are encountered or where the unit is to be used as a radiant heater, the nickel-chrome winding should be specified. The refractory core will not deteriorate, it is claimed, up to 1700 degrees Fahr. A helical groove supports the resistor.



Information supplied by an Industrial Publication

Taylor and White's great discovery which made possible high speed cutting was the greatest single development to help make mass production possible. The steel developed, commonly known as 13-4-1, served industry faithfully for over thirty years without a serious competitor.

In the early 1930's the first molybdenum high speed steels were used on a substantial commercial basis and, before the Second World War, about 25% of all tungsten high speed steels had already been replaced by molybdenum high speed steels on merit. This steady progress of logical replacement of the tungsten steels was interrupted by the war because of the tungsten shortage, when many industrial plants were forced to a sudden change to molybdenum steels.

Because of the stress of war production many concerns have never had an opportunity to satisfy themselves thoroughly as to the comparative merits of the molybdenum high speed steels. For those, in this category, who are inclined to return to the tungsten steels, we suggest a serious consideration of the following facts—molybdenum high speed steels perform as well as, or better than, tungsten steels—and they cost less.

CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

ARMY E NAVY

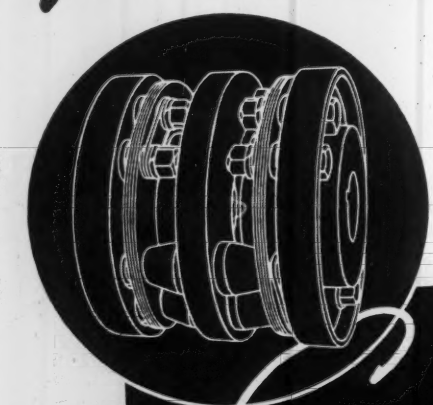
MOLYBDIC OXIDE, BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"

Climax Molybdenum Company

500 Fifth Avenue · New York City

MOLY

THOMAS *Flexible* COUPLINGS



TYPE DBZ

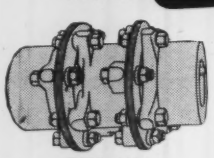
TYPE DSM

TYPE CM

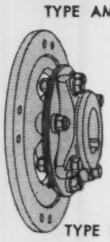
Eliminate:

BACKLASH, FRICTION, WEAR AND CROSS-PULL

the four destructive evils found in other types and makes of couplings.



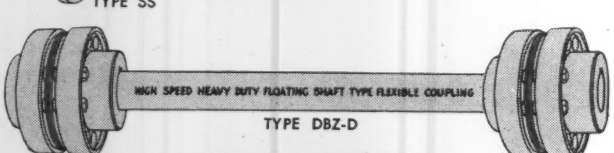
TYPE AM



TYPE SS

NO BACKLASH
NO WEAR
NO LUBRICATION
NO THRUST
FREE END FLOAT

These are the five essential features of Thomas Flexible Couplings that insure a permanent carefree installation.



TYPE DBZ-D

WRITE FOR COMPLETE ENGINEERING CATALOG

THOMAS FLEXIBLE COUPLING CO.
WARREN, PENNSYLVANIA

BUSINESS AND SALES BRIEFS

FORMERLY operations manager of the Los Angeles plant, W. S. Long has been named Pacific Coast sales manager, mechanical goods division, United States Rubber Co. Mr. Long will continue in charge of war products activities on the Pacific Coast.

♦

Shenango-Penn Mold Co., Dover, O., has announced the appointment of Lawrence J. White as sales representative for its centrifugal bronze castings. Previously connected with Automatic Engineering Works which formerly served as agent for Shenango-Penn, Mr. White will have his office at Upton Bank building, Chicago 40.

♦

In charge of sales and development work since 1932, Paul M. Snyder has been named sales manager of Climax Molybdenum Co. and will make his headquarters at Canton, O. John F. Robb, who joined the company in 1935, has been made manager of the Pittsburgh district.

♦

Located at 500 Fifth avenue, New York 18, H. C. Allington has been made sales research engineer of Wickwire Spencer Steel Co. Mr. Allington will be in charge of market development and expansion.

♦

Moving of the Chicago sales office to Suite 341, McJunkin building, 4554 North Broadway, has been announced by Amercoat division, American Pipe & Construction Co., Los Angeles. R. A. Glosgow, general sales manager, and J. W. Staples, Jr., Amercoat sales representative, will continue to serve in the Chicago territory.

♦

Election of John K. Colgate as president of Tyson Bearing Corp., Massillon, O., has been announced. Vice president and treasurer for three years, Mr. Colgate succeeds Ralph Maxson who resigned recently.

♦

Previously commercial engineering manager, Arthur E. Thiessen has been appointed vice president in charge of sales, General Radio Co., Cambridge, Mass.

♦

Opening of a new office at 1505 Commerce building, Houston, Texas, has been announced by Lebanon Steel Foundry, Lebanon, Pa. In charge will be newly appointed southwestern representative Hershel J. Wood, who formerly had been associated with Elastic Stop Nut Corp. of America.

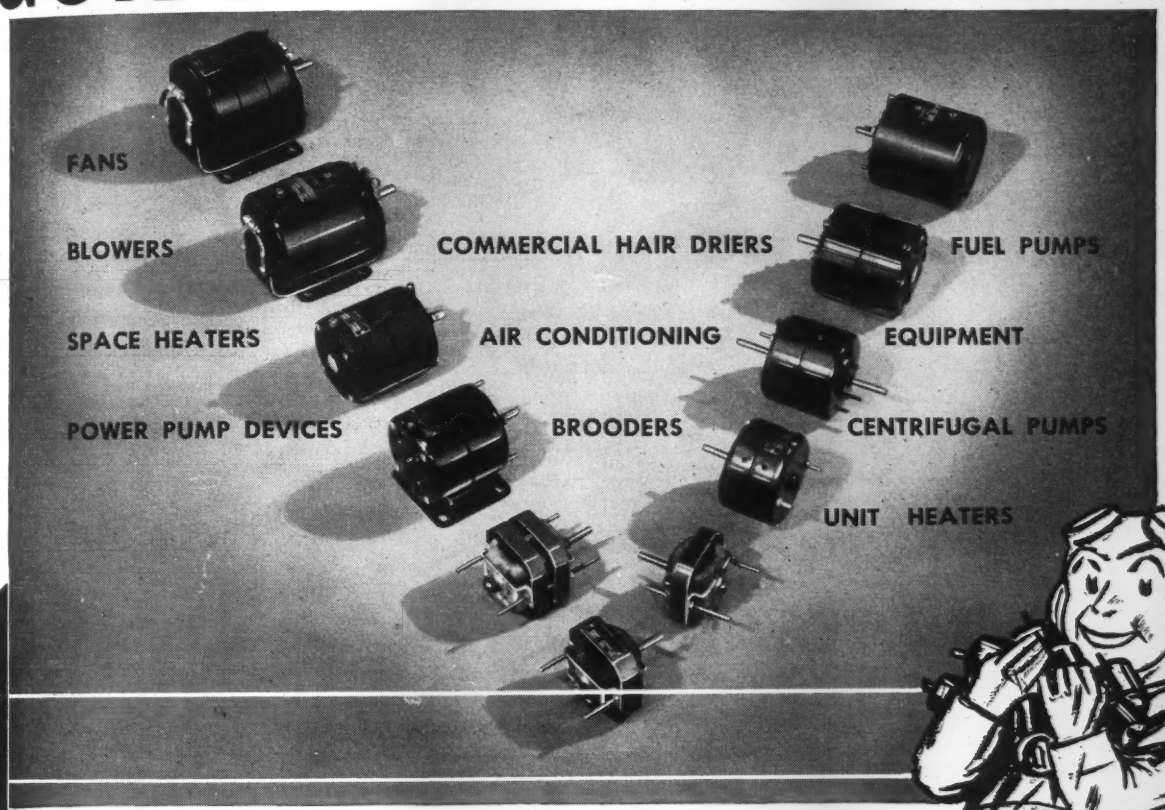
♦

Effective May 1, executive offices of Universal Plastics Corp., New Brunswick, N. J., will be located at Room 2210, 350 Fifth avenue, New York, N. Y. C. W. Marsellus has been appointed director of sales.

♦

Reorganization of personnel in the plastics division has been announced by Monsanto Chemical Co. This includes the appointment of J. C. Brooks as vice president and general manager; J. H. Clark as director of sales; F. A. Abbiati as general

★GUIDE TO POWER DRIVE APPLICATIONS



for that Product on the Board or in Production . . .

. . . if you are designing a product or manufacturing a product now which requires an efficient, low-cost power drive application, it will pay you to investigate the standard line of PILOT Fractional H.P. MOTORS. There is a "job-rated" type to meet practically every power drive requirement . . . available *now* to manufacturers furnishing satisfactory priority ratings.

F. A. SMITH MFG. CO., INC., 401 DAVIS ST., ROCHESTER 5, N. Y.

★SPECIFICATIONS

Frame	Rated H.P.	No Load RPM	Full Load RPM	Length	Diameter	Mounting*	Ask for Bulletin No.
4A	1/20 -1/15	1750	1475	5 5/8"	4 1/16"	R,B,Res.	8A
6J	1/100-1/25	1750	1550	5 1/4"	3 11/16"	R,B,Res.	6J
6J	1/100-1/25	1750	1550	5 1/4"	3 11/16"	R,B,Res.	6J
6H	1/100-1/25	1750	1550	3 3/4"	3 11/16"	R,B,Res.	6H
6G	1/100-1/25	1750	1550	3 3/4"	3 11/16"	R,B,Res.	6G
6A	1/225-1/25	1750	1550	3 3/8"	3 1/16"	R,B,Res.	6 A-1
E2	1/125-1/80	3400	2500	3 7/8"	3 5/16"	R,B	A.K.
S2	1/125-1/80	3400	2500	3 3/8"	3 1/4"		A.K.
E3	1/175-1/125	3400	2500	2 3/4"	3 5/16"	R,B	A.K.
S3	1/175-1/125	3400	2500	2 5/8"	3 1/4"		A.K.
S1	1/500-1/175	3400	2500	2 1/16"	3 1/4"		A.K.

* (R) Plain Round (B) with Base (Res) Resilient Mounting

Write Today for descriptive Bulletins! ----------

Pilot *Fractional H.P.* **MOTORS**

"IF YOU HAVE A TOUGH JOB HANDLING VISCOUS LIQUIDS"

Here's Your Pump

A BLACKMER ROTARY

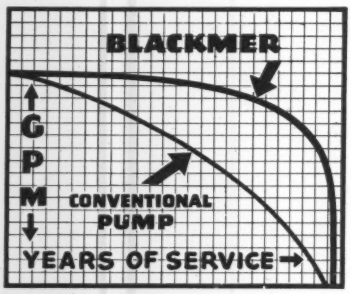
IT IS SELF-ADJUSTING FOR WEAR

4 STANDARD CONSTRUCTIONS
Unlined for oils, syrups, etc.
With removable liners for corrosive or abrasive liquids.
Steam-jacketed for extreme viscosities.
Or with both removable liner and steam-jacket.

A positive displacement pump that handles economically any clean liquid that flows through pipes.

SUSTAINED CAPACITY

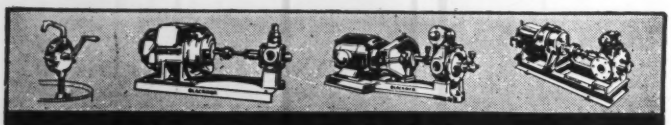
No loss due to internal wear. Bucket Design makes this pump self-adjusting for wear. When the buckets wear out, they are replaced in 20 minutes and the pump restored to its original capacity.



CAPACITIES 5 to 750 GPM—PRESSURES TO 300 psi
Liquid temperatures up to 600° F.
HAND PUMPS MADE IN 54 STANDARD MODELS
Capacities from 1½ to 25 GPM 125 PSI
WE DESIGN AND BUILD SPECIAL PUMPS
BLACKMER NATIONWIDE ENGINEERING SERVICE
awaits your call on problems involving rotary pumps.
BULLETINS FREE TO DESIGN ENGINEERS

No. 304 — Facts about Rotary Pumps
No. 302 — Pump Engineering Data

Blackmer Pump Co., 1974 Century Ave., Grand Rapids 9, Mich.



BLACKMER Rotary PUMPS
"BUCKET DESIGN"—SELF-ADJUSTING FOR WEAR

manager of sales; Charles Lichtenberg as assistant general manager of sales, thermoplastic and thermosetting molding materials and synthetic resins; J. R. Turnbull as assistant general manager of sales, sheet materials; A. C. Martinelli as sales manager, thermoplastic molding materials; S. A. Bell as sales manager, sheet department; and S. L. King as sales manager, vinyl resins department. C. F. Graesser is now in the sales division and Carl Whitlock in the technical service department.

♦

Change of name and address has been announced by Aga Metal Tube Co., Elizabeth, N. J. The company will now be known as Agaloy Tubing Co. and will be located at Springfield, Ohio.

♦

Under the direction of branch manager Roger W. Allen, a new branch office has been opened by General Controls Co., Glendale 1, Calif. The office is located at 376 Nelson street, Atlanta 3.

♦

In charge of the company's new office in the Syndicate Trust building, St. Louis, will be J. W. Workinger who has been recently appointed St. Louis field engineer for Cambridge Wire Cloth Co., Cambridge, Maryland.

♦

Several recent appointments have been made by Pittsburgh Steel Co., Pittsburgh. These are: L. A. Ver Bryck has been appointed New York district sales manager with W. J. Meyer as his assistant, Joseph G. Smith becomes Pittsburgh district sales manager while W. F. Boore has been made assistant sales manager, steel and wire products, at Pittsburgh. General manager of sales since 1942, Joseph A. Voelker has been elected sales vice president.

♦

Formerly chief chemist at Standard Products Co., Port Clinton, O., Roger C. Bascom has joined Hycar Chemical Co. as technical service engineer and will make his headquarters in Milford, Conn.

♦

American Steel & Wire Co. has announced the appointment of W. E. Mackley as manager of manufacturers' sales department of the New York office. Succeeding Mr. Mackley as manager of sales at Buffalo is F. E. Ward.

♦

The territory of Dayton and Cincinnati has been assigned Peninsular Steel Co., Cleveland, by Timken Roller Bearing Co., Steel and Tube Division, Canton 6, O. Heretofore Quality Steels Inc. had covered Dayton, and Frederick Steel Co. had covered Cincinnati.

♦

Appointment of Industrial Equipment Co., 170 Franklin street, Buffalo, as sales representatives has been announced by The Tomkins-Johnson Co., Jackson, Mich. The Buffalo firm formerly had been known as R. C. Neal Co.

♦

Recent announcement has been made by Kiekhaefer Corp., Cedarburg, Wis., of the appointment of John Maypole as Chicago district sales manager. Mr. Maypole had previously been sales engineer.

♦

A new department to be known as the process division has been created by Davison Chemical Corp., Curtis Bay, Md. With field offices placed at various strategic locations, this division will be available to chemical, petroleum, steel and other industries throughout the country for the designing and engineering of process equipment.

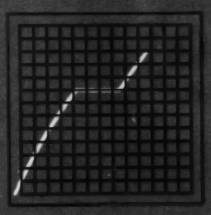
♦

Graham Transmissions Inc., Milwaukee 6, has chosen six more sales engineering organizations to act as representatives. These are: Fitch Bosworth, 7816 Delmar boule-

HIGH SPEED

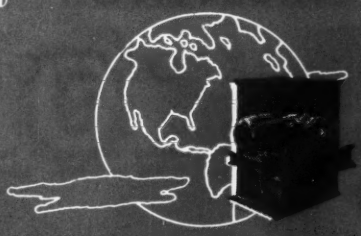
Only with Master Speedrangers can you obtain variable speeds ranging up to 5,000 R.P.M.

You can enjoy all these Variable Speed advantages only with



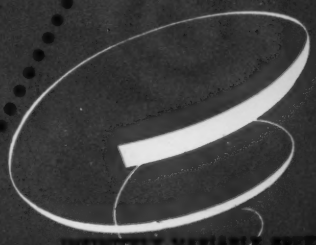
INCREASE PRODUCTION

Use the right speed for each operation and increase production as much as 40%



PROVEN RELIABILITY

Their durability and stamina has been proven with millions of hours of service all over the world.

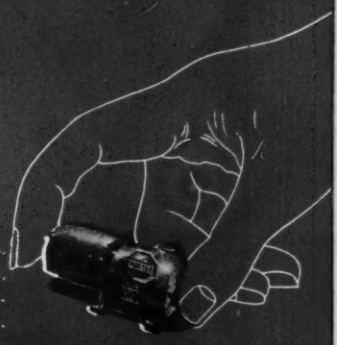


INFINITELY VARIABLE SPEED

Infinite variations in speed may be secured when unit is at rest or in motion.

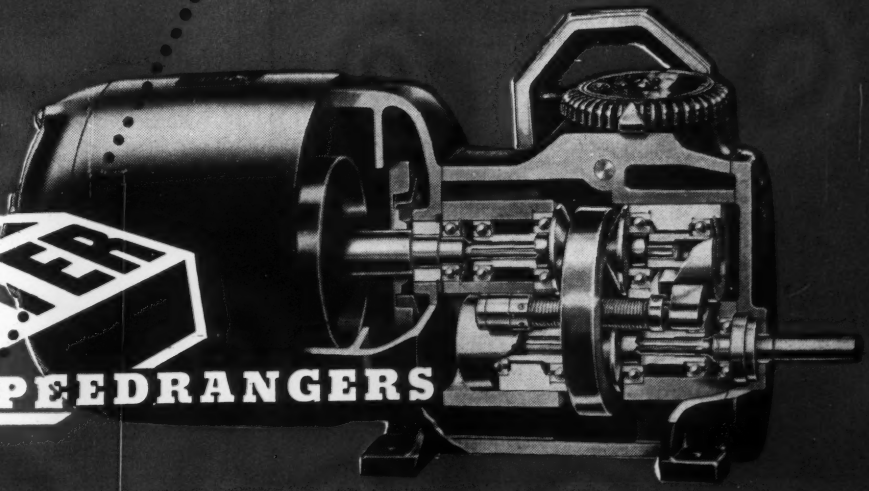
UNDIVIDED RESPONSIBILITY

Complete Speedranger is designed and built by one manufacturer in one plant.



COMPACT

Integral, compact design saves space, saves money, improves appearance.



MASTER SPEEDRANGERS

THE MASTER ELECTRIC COMPANY • DAYTON 1, OHIO